AUGUST 1913 VOL. 35 No. 8

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

PUBLICATION OFFICE, 29 WEST 39TH STREET . . . NEW YORK

CONTENTS

THE JOURNAL is published monthly by The American Society of Mechanical Engineers.
Price, 25 cents a copy, or $2 a year, to members and affiliates of the Society; 35 cents a copy or $3 a year to all others. Postage to Canada, 50 cents additional; to foreign countries, $1 additional.
Entered as second-class matter, January 4, 1912, at the Postoffice, New York, N. Y., under the act of March 3, 1879.

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 35 | AUGUST 1913 | NUMBER 8

The Society's stock of Volume 6 of Transactions is entirely exhausted and to supply urgent demands copies will be purchased at five dollars per volume if delivered at the Society rooms prepaid and in good condition.

THE TRIP THROUGH GERMANY

The tour through Germany of the official party of the Society was fittingly ended on July 9 at Munich, the home of Dr. Oskar von Miller, president of the Verein, and the location of the great German Museum of science and industry, the upbuilding of which has become Dr. von Miller's life work.

The last event at Munich was a banquet given by the city in the old town hall, which was built before Columbus discovered America and which is one of the most beautiful of the old continental town halls.

In its associations and surroundings this event was typical of other receptions and dinners given by the various cities or by the local sections of the Verein deutscher Ingenieure. The first was at Hamburg held in the Rathaus, or Council House. While this is a modern building, completed in 1842, it is noteworthy because of its elaborateness and beauty and is comparable with our largest state capitols, although undoubtedly much finer than any of them in an artistic sense. A similar setting was offered in the Gürzenich at Cologne, built in its original form in 1447 for festivals of the town and the scene of brilliant receptions by

Frederick III, Maximilian I and other emperors of the fifteenth and sixteenth centuries, and in the Römer in Frankfort where the local section of the Verein tendered a luncheon and reception, this being the ancient Rathaus in which were crowned the emperors of the Holy Roman Empire from Charlemagne in 768 to Joseph II in 1792.

The trip included visits to eleven cities, at each of which was accorded a reception far beyond the anticipations of the visitors. In the speeches at the banquets a splendid spirit of coöperation was voiced and a firm conviction of the power of the two great nations represented to continue in amicable industrial relations and to bring about universal peace in the larger affairs of state.

While these functions, with their historically interesting settings, were most impressive and of the utmost importance, the greatest pleasure of the trip came from the gracious hospitality of the German people and the three weeks of intimate association with them. It was very evidently a real pleasure to render the services they did to their American friends and to count them as friends rather than as mere acquaintances. Their hospitality was always genuine and at times overwhelming and it would be difficult to conceive of greater originality than was displayed in the delightful means devised for the entertainment of their guests. Repeatedly some feature was introduced which captivated the American party. Sometimes this was in the form of a musical treat such as a concert by the Leipzig symphony orchestra, singing by the famous boy choir of St. Thomas church in Leipzig, founded by Bach, and the first in Europe; singing by the men's choral society of Cologne, which has won the prize from all Germany for the past two years; the playing of the bugle corps, fifty strong, at Düsseldorf, where the buglers, with instruments hung with flags, played in a stirring manner, raising or lowering their bugles in unison at the beginning or end of each movement; or the dances by the troup of Bavarian peasants brought down from the mountains by the Munich section for their opening event.

Very often, however, the entertainment took the form of an original production applying directly to the occasion. This may have been an original song, or a monologue such as that given in a very pleasing manner by a talented young woman at Cologne just previous to the trip on the Rhine, who personified the Lorelei. The culminating feature of this kind was an allegorical

play at Düsseldorf entitled Der Ring der Arbeit. The piece was elaborately staged with a representation of the interior of a steel works. On one side was the furnace and on the other a huge steam hammer, while in the background were small forges, anvils and other apparatus. In the foreground was the representation of a full-sized ingot which shortly began to glow with heat as fire nymphs from the furnace danced around it. Workmen pushed the ingot under the hammer where it was forged to the accompaniment of an anvil chorus, and when withdrawn and displayed to the audience it had the form of a forged ring, enclosing and holding together the seals of the two societies and the letters V.D.I. and A.S.M.E., with the goddess of liberty appearing close by. Souvenirs were later presented in the form of paper-weights showing in relief the figure of the goddess of liberty and the united seals.

Throughout it was evident that the members of the Verein and their families had given personal attention to many details which contributed to everybody's enjoyment. The special train was met at the station by the local engineers and several times their wives and daughters personally distributed roses to the visiting ladies as they walked up the station platform. A similar courtesy was shown at several of the banquets, and from start to finish, wherever a touch of friendliness could be displayed or personal service rendered, it was sure to be added. This spirit was in evidence first at Plymouth where several German engineers rose at three o'clock in the morning in order to board the boat to accompany the American party to Hamburg, and continued throughout the trip up to the last day at Munich, where the directors of the German Museum presented the Society with a telescope made over 100 years ago by the famous Fraunhofer, the discoverer of the black lines of the spectrum.

The trip through Germany was primarily to observe the engineering and industrial work of the nation, and every opportunity was afforded for the inspection of the leading plants of each city. In every case the reception was cordial in the extreme and often accompanied by elaborate entertainment. Perhaps the most important visits of this kind were the inspection of the inland harbor of Hamburg and the adjacent works of Blohm and Vose, who are now building for the Hamburg-American Line the steamship Vaterland, which is larger than the Imperator. The harbor is 50 miles inland on the river Elbe and excavated almost

entirely from land adjacent to the river banks. It is the largest in Germany and has the third largest shipping trade in the world, with docks to accommodate 450 sea-going vessels and 6400 coasting and river craft. The crane-handling facilities for which the harbor is so well-known were especially interesting, and it is further worthy of note that at a single point were here observed the largest dock, the largest crane and the largest ship in construction in the world.

Scarcely less impressive were the dock facilities inspected still further inland at Duisburg and Ruhrort, constituting the facilities for traffic between the Rhinish-Westphalian coal and industrial district and the waterway of the Rhine. These harbors have a tonnage much greater than any other inland port in the world and only slightly less than that of Hamburg.

No attempt will be made at the present time to refer to the various manufacturing plants visited except to voice the report repeatedly made by the visitors of the splendid welfare and educational work very generally practised by German manufacturers.

As is well known there is a series of insurance laws in Germany requiring provision to be made for the insurance and pensioning of employees and for other features calculated to contribute to their comfort and happiness. Many concerns, however, go far beyond the actual requirements of the law and provide ideal housing arrangements, medical attention, convalescent homes, and factory conditions, which in cleanliness and general attractiveness are almost ideal. The German laws further provide for the instruction of young men who have left school in the intermediate grades, until 18 years of age, usually by means of a night school, and manufacturers now are providing apprentice courses in this connection in which instruction is given in the trades in separate departments and with it the required amount of schooling, but during working hours instead of in the evening.

At Leipzig were the two professional sessions of the Verein deutscher Ingenieure in which The American Society of Mechanical Engineers participated. The first, held in the Central Theater, was a brilliant event attended by the engineers of both nations, occupying the main floor, and the ladies who occupied the balconies. The meeting was honored by the presence of King Friedrich August of Saxony and by many noted men in engi-

neering and science, among them Count Zeppelin. The papers presented were Technical Science and Culture of the Present by Professor Lamprecht, and Engineering Development and Modern Welfare by Dr. W. F. M. Goss, President of The American Society of Mechanical Engineers. At this session honors were conferred upon George Westinghouse.

The second session was held in a hall on the grounds of the Architectural Exhibit now in progress at Leipzig, with papers on Industrial Management by James M. Dodge, Past-President of The American Society of Mechanical Engineers, and on Works Management and Works Theory by Professor Schlesinger. Following this was an opportunity for the inspection of the exhibition, which is on a large scale, with many buildings and numerous model dwellings showing recent ideas in German architecture and house furnishing.

Besides these specially assigned papers, several important lectures were given during the trip by various authorities, notably at Hamburg on the Hamburg harbor; at Leipzig on the Famous Men in Industry, Art, Music and Statecraft who have honored the city; at Berlin on the Relations of the Great Industries of Berlin to those of the United States; at Düsseldorf on the Rhinish-Westphalian Industries; at Duisburg on the Duisburg-Ruhrort Harbor; and at several of the industrial plants visited, notably that of A. Borsig.

Along with the technical features of the journey were several pleasure trips that were welcomed by the visitors as forming a break in their strenuous journey. At Leipzig there was a delightful trip to the Bastai in Saxon-Switzerland on the river Elbe. Here the guests participated in a little mountain-climbing among the high sandstone formations which give the name to this range of hills. From Berlin the party was taken in unusually fine sight-seeing automobiles to the Havel river several miles out, beyond Charlottenburg, where a steamer was boarded for a sail through the picturesque Havel lakes into which the river expands. The steamer was accompanied by a launch carrying a bugle corps alongside to render music, which was much more pleasing than where the band is on the boat itself. A landing was made at the beautiful country-seat of Herr Carl F. von Siemens for participation in a garden party on his spacious grounds, handsomely decorated for the occasion. There was a pleasing entertainment by a choral society from the Siemens and

Halske Works and a surprise in the way of a demonstration of aeroplane flight by a machine owned by the Minister of War. The afternoon spent here was most delightful.

At Heidelberg, the charming old city among the wooded hills, a day's visit was made to the historic castle, the construction of which was begun in the thirteenth century and which was badly ruined during the wars on German soil in the seventeenth century. One of the noted sights is the illumination of this castle at night with colored lights and fireworks, and such an illumination was arranged for this occasion.

The last outing was arranged at Munich and consisted of a trip on the lake of Starnberg. Mention should also be made of the participation in the Fourth of July Celebration, at the Kursaal, Bad Homburg, near Frankfort, in the evening of the Fourth. The view from the Kursaal is particularly beautiful, with large lawn, trees and a distant vista terminating in a wooded slope. The extreme grounds were decorated with lanterns and later illuminated with fireworks. The celebration included a dinner and was under the auspices of this Society.

In Leipzig, Berlin, Düsseldorf, Cologne, Frankfort and Mannheim special automobile trips were planned for the ladies of the party. In almost every city private cars were loaned and three or four American women were put in the charge of one German woman. In Leipzig the ride ended in the exhibition grounds; in Berlin it included a visit to the palace, Charlottenburg and the mausoleum of Queen Luise, and a delightful luncheon at the Grunewald Rennbahn; in Düsseldorf the most interesting part of the drive was the visit to the Friedenskirche where not only the beautiful frescoes of Gebhardt were to be seen, but the artist himself, who explained his paintings in person. During the Cologne drive another old church was visited, St. Gereon's, and the American women were entertained at luncheon.

The women of Frankfort gave their guests an interesting drive to the Saalburg, a wonderfully restored old Roman fortress. The Mannheim women allowed the American women the privilege of visiting several of their own homes and entertained them delightfully, in the last one with tea, music and dancing in the natural theatre in the gardens. Besides these automobile trips, the American women were given the opportunity of inspecting the institutions for the welfare of the workmen of the Friedr. Krupp Company at Essen and of visiting various museums and invaluable art treasures in the different cities.

In closing, the German Museum at Munich should be mentioned again, for nothing on the trip was more remarkable than this and in the estimation of many it takes precedence over anything seen elsewhere. It was founded only ten years ago and already occupies two large buildings with a wonderfully complete collection of demonstrating models representing the development of astronomy, physics, chemistry, metallurgy, mechanics, shipping, transportation by rail, and the various branches of the sciences and arts. A magnificent new building is now nearly completed for the housing of these exhibits under one roof and the visitors of this Society had the pleasure of lunching there and of presenting to the Museum at that time a large model of the Panama Canal.

At the final dinner at Munich both Dr. von Miller, President of the Verein deutscher Ingenieure and Dr. C. Matschoss, who has personally attended to the management of the visit of the American engineers, received an ovation. Everyone was enthusiastic in his praises of the splendid hospitality which had been shown on every hand. It seemed impossible to express the deep feelings of appreciation of which all were sensible and the wish was expressed, as it had been on previous occasions during the trip, that the German engineers might come to America in 1915 to attend the engineering congress in San Francisco at the time of the opening of the Panama Canal and give their American friends an opportunity to render them some service in return.

A complete account of the German trip and of the meeting in Leipzig will be published by the Society at a later date.

KELVIN MEMORIAL WINDOW

The memorial window in honor of Lord Kelvin, erected in Westminster Abbey by the engineering societies of Europe and America, was unveiled on the afternoon of July 15 and appropriately dedicated by the Dean of Westminster, Bishop Ryle, in the presence of a large congregation. The Society was represented at this ceremony by about forty members.

The window, which has been placed on the north side of the nave, close to the one erected in 1909 to the memory of Sir Benjamin Baker, was designed and carried out by the same artist, Mr. J. N. Comper. It overlooks the grave of Lord Kelvin, and the lights contain two large figures under canopies, King Henry V in armor, and Abbot William Colchester vested as in effigy on his tomb in the abbey. Within the canopy above the king is a picture of his coronation and above the abbot a representation of the contemporary story of King Henry V visiting the Westminster recluse on the night after his father's death. Both lights are framed with borders having niches holding sixteen statuettes and thirteen shields relating to them. In front of the pedestals of the two large figures are tablets held by angels, containing the words, In Memory of Baron Kelvin of Largs, Engineer, Natural Philosopher, B.1824, D.1907, and beneath, the arms of Lord Kelvin and of Glasgow University.

Preceding the dedication the Dean of Westminster paid a short tribute to Lord Kelvin, dwelling upon the extraordinary group of Cambridge men with which he was associated, all of whom had not only a good head but a good inheritance. It had been the part of Kelvin to transform to the practical utility of mankind the glories of the new fields of knowledge opened up in the Victorian era and the new world of electricity which had been discovered. He was never spoilt by fortune in the pursuit of his researches and never allowed himself to be a recluse; he was always genial and accessible, always humble and unselfish. Nothing was too simple for his experiments, nothing too abstruse for the powers of his calculation. All through his life, in the

face of the strong prevailing current of materialism, he preserved the simplicity of his early Christian faith. He spoke with humility of a great man and many could look back with gratitude upon the influence of the example of his religious belief which a man of his gigantic intellect furnished to those of a younger generation. One of the inventions which his genius succeeded in perfecting was that for submarine telegraphy across the Atlantic, and Englishmen and their American brothers had thereby been brought into immediate, constant and almost instantaneous communication, and a sense of brotherhood, whose peace had been unbroken for one hundred years and which it was to be hoped would so continue, had been materially deepened and strengthened. Kelvin's name on both sides of the Atlantic was one of the most epoch-making in the domain of natural philosophy.

A procession was then formed to the nave where the Dean signified his acceptance of the window as part of the fabric of the church. The window was presented to the Abbey by Mr. R. Elliott Cooper on behalf of the donors.

After the service the Dean of Westminster and Mrs. Ryle received the engineers in their home, and a large number of the American party participated in this courtesy.

In the evening, the societies in London jointly invited the American engineers and their ladies to a fête in the Royal Botanical Society's Gardens in Regents Park. Here a reception was held in the beautifully illuminated gardens and a performance of King René's Daughter was given on the greensward amidst natural scenery. Supper was then served in the marquee.

Mr. Cooper welcomed the guests on the part of the hosts and generously gave credit for the part which the American societies had taken in the erection of the window. In the absence of both Mr. Gates and Mr. Dean, who had expected to be present, Calvin W. Rice, Secretary of the Society, responded and acknowledged the further obligation of the members for the hospitality of their English friends. He warmly invited all to come to America in 1915 to permit some suitable return of entertainment. He spoke also of the important part the engineer is taking in the world's work and said that the bringing together of the members of the profession was instrumental in bringing together the nations.

APPLICATIONS FOR MEMBERSHIP

The Membership Committee have received applications from the following candidates. Any member objecting to the election of any of these candidates should inform the Secretary before September 15, 1913:

Ames, John McEwen, New York, N. Y.
Bailey, Charles Edward, Chambersburg, Pa.
Bell, Andrew Lane, Culebra, C. Z.
Brown, Stuart F., Whitinsville, Mass.
Cajander, Sven Gottfrid, Chile, S. A.
Chadwick, George Albert, Alexandria, Va.
Clarke, Walter J., San Francisco, Cal.
Conti, Angelo, Washington, D. C.
Cook, Charles Bannister, Hartford, Conn.
Courtenay, Charles R., Watertown, N. Y.
Fortney, Camden Page, Gatun, C. Z.
Frick, John, Philadelphia, Pa.
Gibson, Frederic Arthur, Huddersfield, England
Haines, Philip G., Rochester, N. Y.
Hardesty, Frederick Sawyer, Washington, D. C.
Henley, Wm. Wheeler, Tucson, Ariz.
Hogg, James, Amsterdam, N. Y.
Hopkinson, Joseph, Dayton, Ohio
Kugel, H. K., New York, N. Y.
Lynn, Thomas H., Williamsport, Pa.
Menzin, Abraham Louis, Edge Moore, Del.
Oram, Robert C., Los Angeles, Cal.
Parker, Francis Warner, Jr., Chicago, Ill.
Schultz, John L., Philadelphia, Pa.
Seligman, Walter, New York, N. Y.
Silbert, S. Joseph, Plainfield, N. J.
Smith, Percy Chas., Hartford, Conn.
Spicer, Clarence Winfrid, Plainfield, N. J.
Spink, Arthur J., Detroit, Mich.
Stillman, Thomas B., Jr., Hoboken, N. J.
Taylor, William T., Lancashire, England
Thomas, Walter Henry, Sheffield, England
Trabold, Frank William, Brooklyn, N. Y.
Trinder, Frederick J., New Britain, Conn.
Uihlein, Herman A., Racine, Wis.
Walter, William Edward, Rutherford, N. J.
Weigel, Albert C., Chattanooga, Tenn.
Wesselhoeft, Charles Dietrich, Chicago, Ill.
Wischmeyer, Carl, Terre Haute, Ind.
Wood, Stanley Victor, New York, N. Y.

PROMOTION FROM ASSOCIATE

Chapman, David A., Winthrop, Mass.

PROMOTION FROM JUNIOR

Braun, Carl F., San Francisco, Cal.
Bushnell, Leonard T., Seattle, Wash.
Hatch, Edwin Glentworth, New York, N. Y.
Houskeeper, Wm. Gibbons, Philadelphia, Pa.
Miller, Edward G., Juncos, P. R.
Miller, Elmo J., Humacao, P. R.
Millett, Kenneth Ballard, Willimantic, Conn.
Robinson, Rhea Hamilton, Milwaukee, Wis.
Sar Vant, Wilbur Nason, Brooklyn, N. Y.
Woodruff, Clarence Atha, Bridgeport, Conn.

SUMMARY

New applications	40
Promotion from Associate	1
Promotion from Junior	10
	51

LIBRARY GIFTS

The Library Committee is considering seriously the establishment of branch libraries in several cities in the United States outside of New York, or at least the strengthening of the engineering resources of public libraries existing in these cities. Many members of the founder societies have been generous in donations of sets of periodicals to the Library, and thus there have accumulated a number of duplicate sets of periodicals. It is hoped that any member of the Society who has such sets which he may care to donate for this purpose will communicate with Dr. Leonard Waldo, at 49 Wall Street, New York, stating his intention, and submitting a list of the material which he may wish to present. Periodical sets and sets of society transactions are especially welcome.

The Library has recently received a most generous gift of a hundred or more books from Mrs. Charles Wallace Hunt, widow of the Society's past-president, which have been placed in the Library's stacks where they are accessible to readers. The list is too long to be included in the Accessions to the Library, published on another page of The Journal, but it comprises many volumes of technical value and forms a distinct addition to the Library's valuable collection.

PITOT TUBES FOR GAS MEASUREMENT

By W. C. Rowse

ABSTRACT OF PAPER

A series of experiments has been made by the author at the laboratories of the University of Wisconsin to obtain information concerning the reliability of the pitot tube as a means of measuring gases and to determine the accuracy of various forms of the instrument which are in common use. All tubes were compared with a Thomas electric gas meter which was taken as a standard of measurement. Since any variation in results would be due to a wrong method of obtaining the static pressure, simultaneous readings were taken of velocity heads as shown by the pitot tube using the pitot static pressure and by the pitot dynamic tube and a piezometer. The pressure as obtained by the piezometer would not be affected by the form of tube used.

The results, which are treated in Par. 42 to 46, may be discussed briefly as follows:

a The pitot tube is a reliable means of measuring gases when the static pressure is obtained in a correct manner and when all readings are taken with a sufficient degree of refinement.

b The piezometer is the most reliable means of obtaining the static pressure.

c Of the various forms of static openings in the pitot tube itself, very small holes in a perfectly smooth surface give the most accurate results.

d Slots give erroneous static pressures and beveled-end tubes for obtaining static pressures are not reliable.

e The German Stauscheibe is a reliable means of measuring gases.

PITOT TUBES FOR GAS MEASUREMENT

By W. C. Rowse, New York

Junior Member of the Society

The measurement of gases is receiving increased attention in this country and in Europe. The vast quantities of natural and manufactured gas consumed for power, lighting and heating must be measured with precision not only because of their value but because modern business methods demand accuracy. Blowers and ventilating fans are usually sold under a guarantee that they will deliver a certain volume of air or gas under given conditions, but often the two different methods employed by the seller and the purchaser in measurement disagree and neither one has indisputable evidence that his method is correct. Other instances might be cited to show the need of more knowledge concerning the accuracy of various methods of measuring gases.

2 The pitot tube as a means of measuring gases has been described by many writers and investigators.[1] It has the advantage of being correct in principle, inexpensive, portable, and is in general easily applied. But the accuracy of the different forms of pitot tubes has long been questioned, the maker of each form supposing that his tube is correct, while as a matter of fact no two forms of tube agree. There has long been need for a careful, scientific study of the pitot tube for the measurement of gas and a fair comparison of the different forms in common use.

3 The purpose of the experiments which were made in the

[1] D. W. Taylor, Experiments with Ventilating Fans and Pipes. Society of Naval Architects and Marine Engineers, November 1905; Frank H. Kneeland, Trans. Am. Soc. M. E., vol. 33, p. 1137; G. F. Gebhardt, Journal Am. Soc. M. E., November 1909; R. Burnham, Engineering News, December 21, 1905; Forrest M. Towl, Columbia University Lectures, 1911; Chas. H. Treat, Trans. Am. Soc. M. E., vol. 34, p. 1019; Thos. R. Weymouth, Trans. Am. Soc. M. E., vol. 34, p. 1094.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

laboratories of the University of Wisconsin and which form the subject matter of this paper may be stated briefly as follows:

a To investigate the reliability of the pitot tube as a means of measuring gases.

b To ascertain which forms of the pitot tube now in common use give correct and which incorrect results in the measurement of gases.

4 It was planned to force air through a pipe in which the pitot tube to be tested was inserted, together with a standard

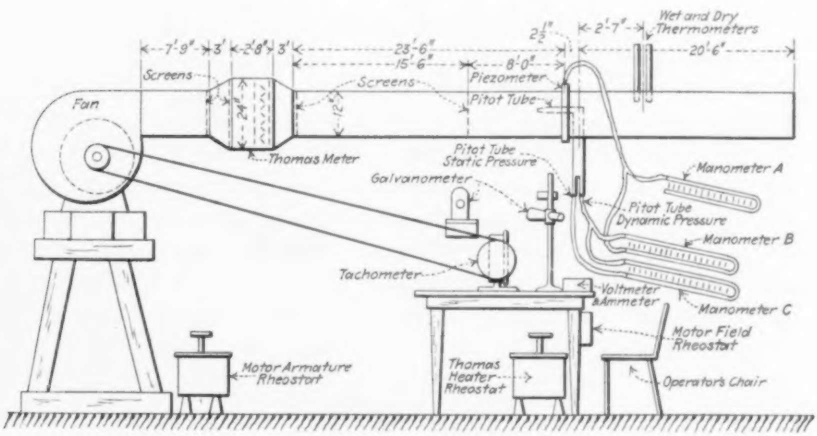


Fig. 1 Sketch of Apparatus

gas meter, known to be correct. This plan as a whole presented no difficulties except the selection of a standard gas meter concerning whose accuracy there could be no question.

5 The only fundamentally correct means of measuring directly the volume of a large quantity of flowing gas is the displacement or "holder" method. A holder of known dimensions drops a given distance, thus forcing a certain definite volume of gas through the discharge pipe, where it is again measured by the meter to be tested. The temperature, pressure and humidity of the gas both in the holder and at the apparatus to be tested must be known in order that both volumes may be reduced to the same conditions and thus a fair comparison may be made. It is a very difficult matter to obtain fair average readings of these quantities, especially of the temperature, because of the influence of the water in the holder, of the weather conditions outside, and of the large volume of gas in the holder. There

are certain periods in the spring and autumn when the temperature remains practically constant day and night for several days, and this is the only time when holder tests can be made with any approach to accuracy.

6 These considerations, as well as the fact that the holder is clumsy, intermittent in action and altogether unsuited for laboratory experiments, made its use as a standard of measurement out of the question.

7 The Thomas electric gas meter [1] which has been developed

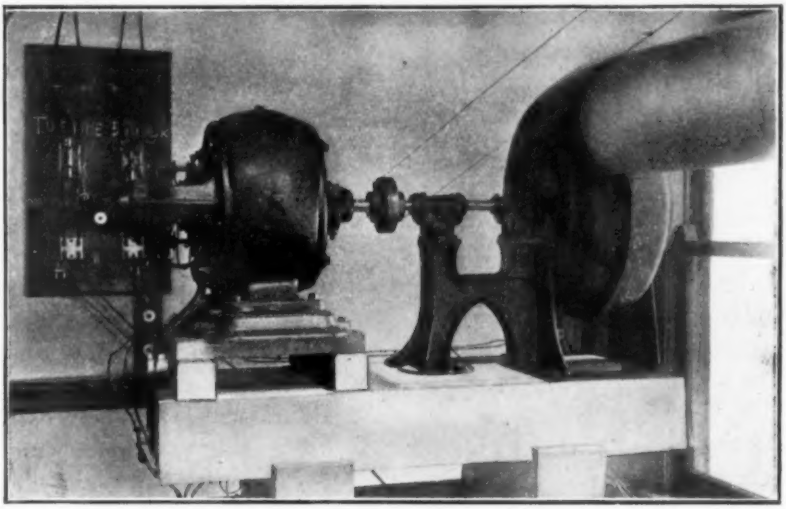


Fig. 2 View of Fan and Motor

during the last few years and which is described in detail in a later portion of this paper, was used as the standard gas meter in the experiments with pitot tubes for the following reasons:

a It measures weight of gas, and avoids the difficulties inherent in volumetric measurements.

b Its measurement of gas depends directly upon the specific heat of a gas. The specific heat varies but slightly with wide changes in temperature, pressure and humidity.

[1] Carl C. Thomas: Electric Gas Meter, The Journal, Am. Soc. M. E., December 1909; Measurement of Gas, Journal of the Franklin Institute, November 1911; Some Recent Developments in Gas Measuring Apparatus Proc., Am. Gas Inst., vol. 7, 1912.

c Its accuracy is limited only by the exactness of electrical measurements; such measurements can be made by engineers with a very great degree of refinement with well-known highly developed instruments.

d It has been thoroughly tested by experiments under the most varied conditions in this country and abroad; some of the results of tests are presented in Appendix No. 1, and in every instance it was proven that the meter was correct not only in theory but also in the actual measurement of gases.

e Its operation is very simple, the readings are few and can be obtained with the greatest accuracy, while it requires almost no attention itself when in use.

DESCRIPTION OF THE APPARATUS

8 The apparatus used in the experiments on pitot tubes is shown diagrammatically in Fig. 1 and by photographs in Figs. 2, 3 and 4.

9 A No. $2\frac{1}{2}$ Sirocco fan driven by a $7\frac{1}{2}$-h. p. direct-current shunt-wound motor forces air through the Thomas meter into a galvanized iron pipe 12 in. in diameter in which the pitot tube to be tested is inserted.

10 Variable resistances were placed in series with both the field and the armature of the motor, thus making possible a wide variation in speed. A stationary tachometer belted to the fan was so located that it could be observed at all times by the experimenter at the pitot tube. The field rheostat was brought within reach of the operator so that the fan could be maintained at any desired constant speed during a test.

11 Screens were inserted at the points shown in Fig. 1 in order to break up eddies and whirls and to have the air flow as nearly parallel as possible at the point where the pitot tube readings were taken. All joints between the Thomas meter and the pitot tube were made thoroughly air tight to prevent leakage. The barrel of the Thomas meter was lagged by three thicknesses of heavy blanket in order to prevent any possibility of error due to radiation to or from the meter casing.

12 The experimenter was stationed directly under the pitot tube and all readings were taken at this point. A mercury barometer, hung on the adjacent wall, gave the atmospheric pressure, and a manometer inclined at a 10 to 1 slope made it possible to

determine accurately the static pressure in the pipe above atmosphere. The dry-bulb thermometer indicated the temperature of the air flowing in the pipe and together with the wet-bulb thermometer gave readings from which the humidity could be determined.

THE THOMAS ELECTRIC METER

13 The Thomas electric meter is based on the principle of

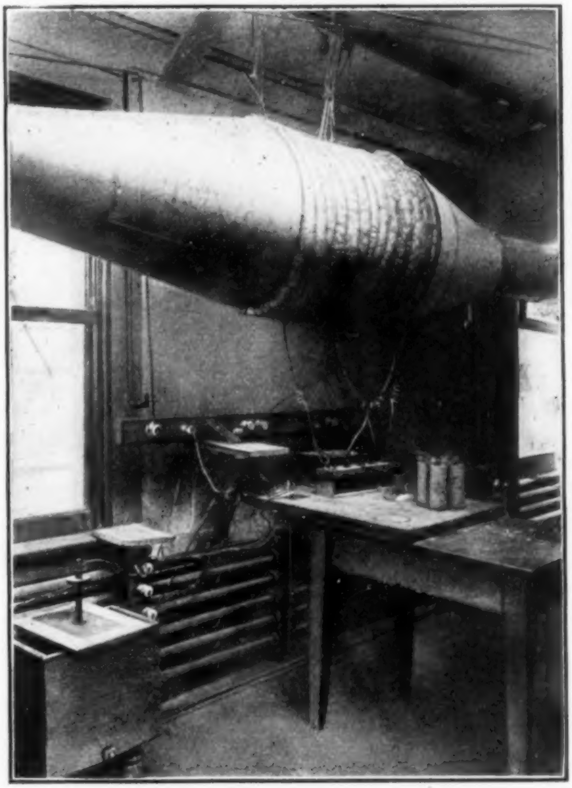


FIG. 3 VIEW OF THOMAS METER

heating the gas through a known range of temperature and measuring the energy required to cause this change in temperature: this measured heat is proportional to the weight of gas flowing. Electric energy is used as the source of heat as it can be accurately measured and easily controlled. The temperature range is determined by the use of electrical resistance thermometers.

If E is the amount of energy in watts supplied to the heating coils to raise the temperature of W lb. of gas per minute through t deg. fahr., and if s is the specific heat at constant pressure of 1 lb. of gas, then

$$W = \frac{0.05686E}{ts}$$

14 The manually controlled Thomas meter used in these experiments is shown diagrammatically in Fig. 5. An electric

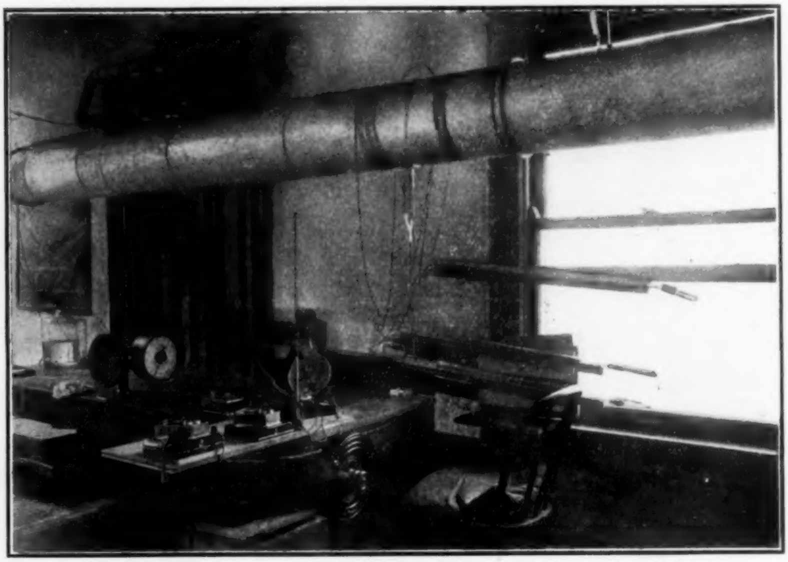


FIG. 4 VIEW TAKEN AT OBSERVER'S STATION

heater is placed within a casing between two electric resistance thermometers T_1 and T_2. The heater consists of bare resistance wire mounted on a fiber frame and evenly distributed over the section of the casing. A water rheostat is placed in the heater circuit for regulating the direct-current supplied. The energy is measured by an ammeter and a voltmeter, both accurately calibrated in the standards laboratory of the electrical department of the University of Wisconsin after each series of tests.

15 The thermometers consist of nickel resistance wire wound on insulated wooden spindles which are evenly distributed over the cross-section of the casing. These thermometers were calibrated simultaneously by the author by means of a special ap-

paratus at the plant of The Cutler-Hammer Manufacturing Company, Milwaukee, Wis., where the commercial form of the Thomas electric meter is manufactured. The results of this calibration are given by the curves in Fig. 26, Appendix No. 2. This curve shows the ohms increase in resistance per degree rise in temperature for any ordinary temperature. These two thermometers were arranged to form two arms of a Wheatstone bridge,

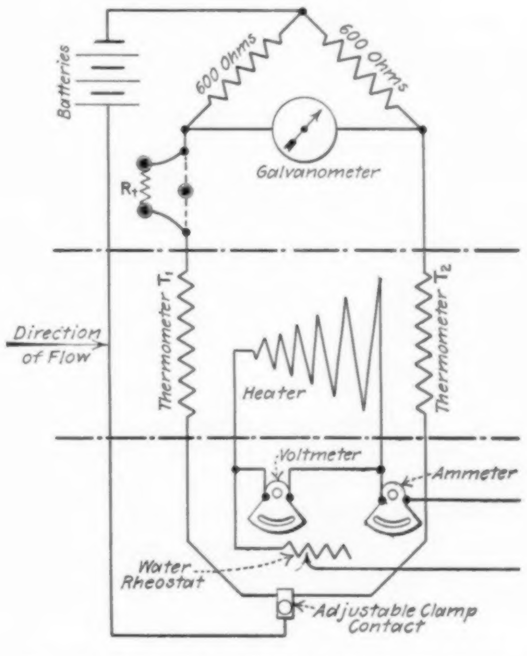


FIG. 5 DIAGRAM OF THOMAS ELECTRIC METER

of which the other two arms were fixed resistance coils (600 ohms each), made of wire having a zero temperature coefficient. A galvanometer is connected across the Wheatstone bridge thus formed and an adjustable clamp contact is provided to balance the bridge when no heat is passing through the heater. A small resistance R_t, also with zero temperature coefficient, is arranged so that it can be placed in or out of series with the entrance thermometer. The resistance R_t used in these experiments was 2.7998 ohms at all temperatures.

16 The operation of the meter is as follows: With gas flow-

ing steadily through the meter but with no energy in the heater, and with R_t out of circuit, the Wheatstone bridge is balanced by means of the adjustable clamp contact. Then the resistance R_t is put in circuit and sufficient electrical energy is supplied to the heater to bring the galvanometer to a balance again. This balanced state can be attained only by bringing the exit gas to such a temperature above that of the entering gas that the resistance of thermometer T_2 has increased 2.7998 ohms above the resistance of the thermometer T_1. The required increase in temperature of the gas varies slightly with the average gas temperature and may be found from the curve in Fig. 26, Appendix No. 2. The electrical measuring instruments in the heater circuit indicate the energy input which has been required to raise the temperature of the gas through the known range. The pounds of gas flowing per minute can then be found by the equation given in the preceding description.

17 The experimental work to be described was done after about two years' experience with the Thomas meter as a piece of regular laboratory apparatus, during which time its accuracy had been fully established.

THE PITOT TUBES

18 The pitot tube is a well-known measuring instrument and needs only a brief description. It consists essentially of two parts: a dynamic tube pointing upstream which converts the sum of the pressure energy and the velocity energy into a head which may be measured; and a means of determining the pressure head (or static pressure) alone. The difference between the dynamic head and the static pressure head is the velocity head h in the fundamental formula for the flow of fluids

$$v = \sqrt{2\ gh}$$

where

v = velocity in ft. per second

g = 32.2 ft. per second per second

h = mean velocity head in ft. of the fluid flowing

19 It has been satisfactorily proved and accepted that the dynamic tube gives the correct pressure if the tube points parallel to the current. But it is a very difficult matter to obtain the correct static pressure on account of secondary velocity effects. Therefore the study of the accuracy of the pitot tube resolves

itself into a study of the correct method of obtaining the static pressure at the given cross-section where the tube is inserted.

20 Each pitot tube tested had as a part of the tube a means of determining the static pressure, and readings of the velocity head were obtained by using this pitot tube static pressure. Simultaneous readings of the velocity head were obtained by using a piezometer ring for the static pressure together with the dynamic tube of the pitot tube under test. This is further illustrated by reference to Fig. 1. Manometer *B* gives the velocity head by using the piezometer static pressure, and manometer *C* gives the velocity head by using the pitot tube static pressure.

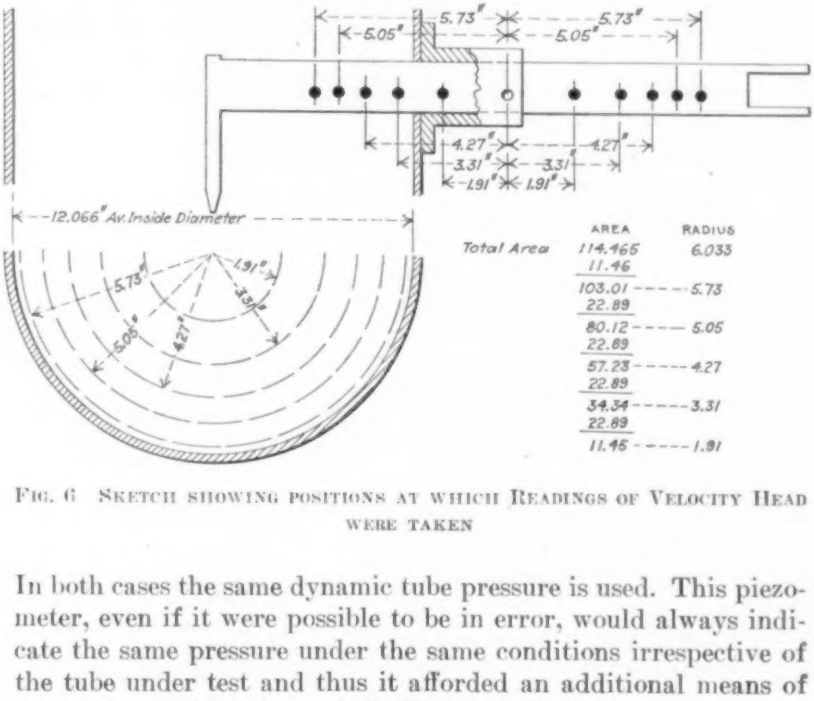


FIG. 6 SKETCH SHOWING POSITIONS AT WHICH READINGS OF VELOCITY HEAD WERE TAKEN

In both cases the same dynamic tube pressure is used. This piezometer, even if it were possible to be in error, would always indicate the same pressure under the same conditions irrespective of the tube under test and thus it afforded an additional means of *comparison* independent of the Thomas meter.

21 The piezometer is shown in Fig. 1 and Fig. 7 and is simply an absolutely air-tight annular space about the pipe, connected with the interior of the pipe by six small holes 0.04 in. in diameter.

22 The velocity of a gas flowing through a pipe is much greater at the center than near the walls of the pipe.[1] In addi-

[1] Loeb, Journal American Society of Naval Engineers, 1912, p. 1115.

tion, it was apparent from the tests that the gas flows through the pipe with a wave or spiral motion even when many screens are inserted to straighten out the stream lines. Therefore, it was

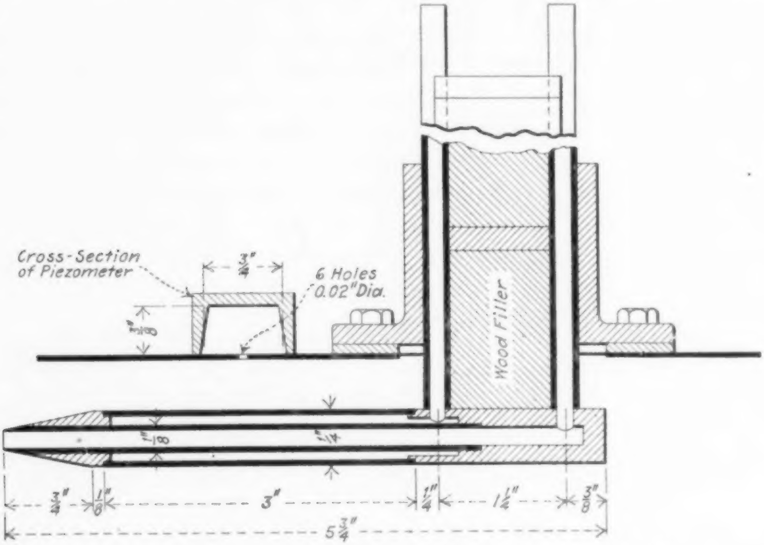


FIG. 7 DIMENSIONED SKETCH OF PITOT TUBES *A* TO *H*

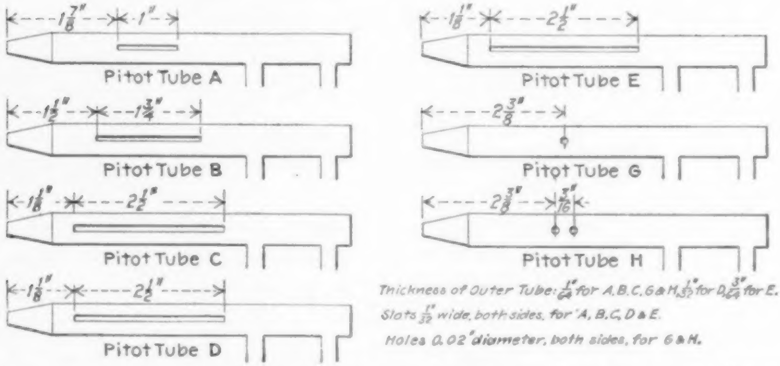


FIG. 8 SKETCHES OF PITOT TUBES *A* TO *H* SHOWING DIMENSIONS OF STATIC OPENINGS

necessary to take a large number of readings across two diameters of the pipe. The total area of the pipe was divided into five concentric annular areas and four readings of the velocity head were obtained in each test at the center of each annular area, thus giving 20 readings from which the mean velocity head

could be calculated. Since the velocity varies as the square root of the velocity head it was necessary to average the square roots of each of the 20 readings, and the square of this average represented the mean velocity head. The positions on the diameter at which the readings were taken are shown in Fig. 6.

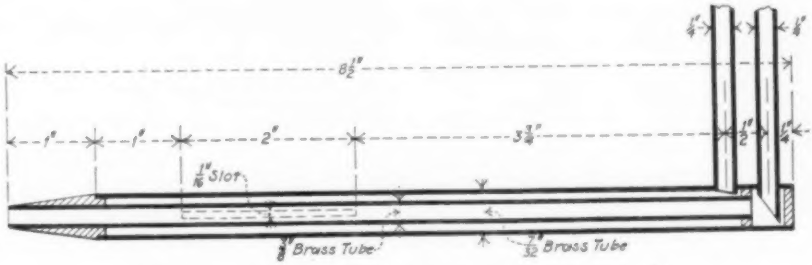


FIG. 9 DIMENSIONED SKETCH OF PITOT TUBE X

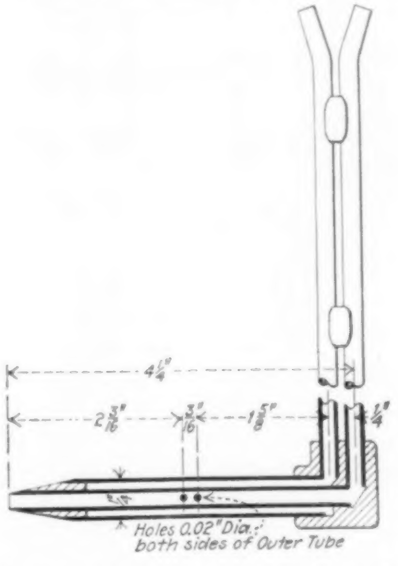


FIG. 10 DIMENSIONED SKETCH OF PITOT TUBE Y

Readings were also taken at the center of the pipe in order to determine if any definite relation existed between the mean velocity head and the velocity head at the center of the pipe.

22 The pitot tube under test was held in place by means of a brass bushing or holder which could be fastened by screws to brass facings soldered to the outside of the galvanized air

pipe. The facings were 90 deg. apart, thus permitting readings to be taken on both the horizontal and vertical diameters. Each pitot tube was first carefully centered in the pipe and a dowel hole bored in the shank corresponding to a hole in the holder. Ten other holes were bored in the shank at the proper distances either side of the center (see Fig. 6), so that by the use of a dowel pin the pitot tube could be quickly and accurately placed in its proper positions in the tube.

23 The pitot tubes tested in these experiments may be de-

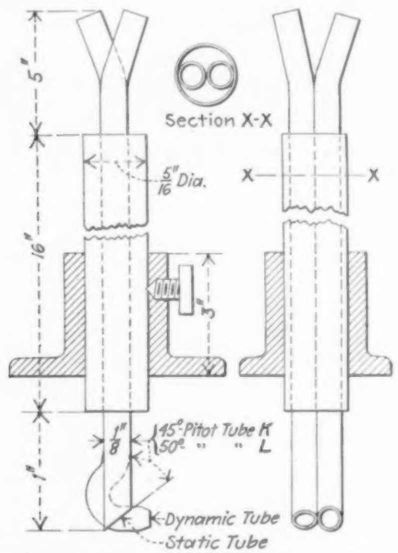


Fig. 11 Dimensioned Sketch of Pitot Tubes K and L

scribed briefly as follows: Tubes A, B, C, D, E, G and H are shown by sketches in Figs. 7 and 8. An experimental pitot tube was constructed as shown in Fig. 7 with a removable outer tube which contains the static opening. By inserting different outer tubes the seven pitot tubes shown in Fig. 8 were made. Tubes A, B and C are alike except for the length of the static slot. Tubes D and E are like tube C except for the thickness of the outer tube as shown in Fig. 8. Tubes G and H have small holes 0.02 in. in diameter for the static openings.

24 Tubes X and Y were loaned to the University of Wisconsin through the courtesy of Mr. F. R. Still of the American Blower Company. Tube X, Fig. 9, was made from a drawing

furnished by Captain D. W. Taylor,[1] U. S. N.; this form of tube is used as a standard for ventilation work by the United States Navy. Tube Y, Fig. 10, is the standard tube of the American Blower Company, and was developed by Mr. Chas. H. Treat, who has thoroughly tested the tube for accuracy.[2]

25 Tubes K and L, Fig. 11, were constructed from descriptions of the tubes developed by Prof. G. D. Gebhardt[3] of the Armour Institute of Technology. These were alike except that the static opening of the tube K was beveled to 45 deg., while in tube L, the static opening was flatter, being beveled to an angle of 50 deg. as shown in Fig. 11.

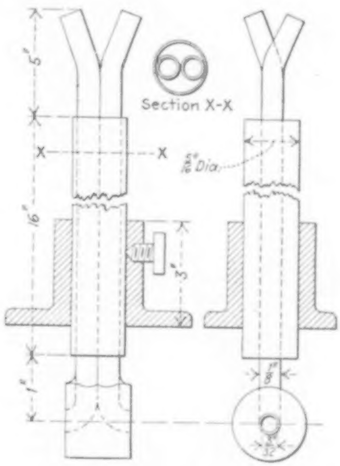


FIG. 12 DIMENSIONED SKETCH OF GERMAN STAUSCHEIBE

26 The "Stauscheibe,"[4] Fig. 12, has been widely used for several years in Germany and the experiments on this early form of pitot tube will be of interest for purposes of comparison.

GAGES FOR MEASURING THE VELOCITY HEADS

27 The velocity heads measured were very small quantities, ranging from 0.08 in. to 1.6 in. of gasolene, so that gages of

[1] Capt. D. W. Taylor; Soc. Naval Architects & Marine Engrs., Nov. 1905.

[2] Chas. H. Treat; Measurement of Air in Fan Work, Trans. Am. Soc. M. E., Vol. 34, p. 1019.

[3] Trans. Am. Soc. M. E., Vol. 31, p. 601.

[4] Reitschel; Versuche uber den Widerstand bei Bewegung der Luft in Rohrleitungen. Gesundheits Ingenieur, Festnummer July 1905. Marx; Uber die Messung von Luftgeschwindigkeiten. Gesundheits Ingenieur 1904.

unusual accuracy had to be used. From previous experience it was known that the inclined manometer when properly constructed and carefully calibrated would give readings which were correct within the required limits of error. Two gages were therefore constructed as shown by sketch in Fig. 13. The glass tubes were approximately 0.575 in. outside diameter and were selected with the greatest care from a large stock, special atten-

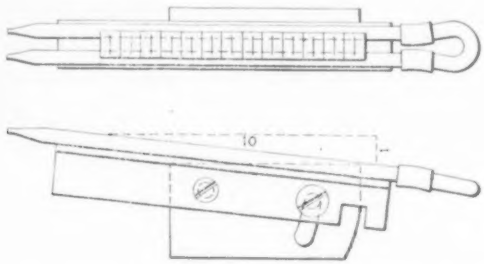


FIG. 13 SKETCH OF INCLINED MANOMETERS *A* AND *B*

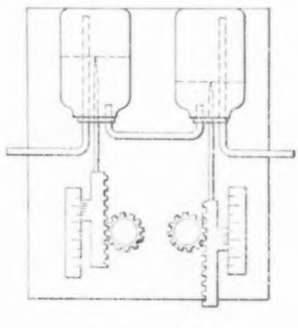


FIG. 14 DIFFERENTIAL HOOK GAGE USED AS A STANDARD

tion being given to straightness, uniformity of bore and freedom from flaws. These gages were placed at a ten to one slope by means of an accurate template and a spirit level which was correct to 0.002 in. in 10 in. They were calibrated before and after each series of runs by means of the differential hook gage and micrometer, shown in Fig. 14. A few such representative calibrations are given in the Table 1.

28 A sliding scale made it possible to read the velocity head directly. Gasolene was used as a manometer fluid because it automatically kept the inside of the glass tubes clean, had a very

definite meniscus and had almost no capillary attraction for the glass. Several preliminary tests were run to determine whether the vapor tension of the gasolene vapor affected the readings. The manometers were first made to check each other when both contained gasolene and then the same pressure was measured when one manometer was filled with gasolene and the other with kerosene. Identical pressure readings were obtained with kerosene as with gasolene. It, therefore, seemed evident that the

TABLE 1 SAMPLE CALIBRATIONS OF MANOMETERS *B* AND *C*

PRESSURES IN INCHES OF GASOLENE

Differential Hook Gage	Manometer *C*	Manometer *B*	Differential Hook Gage	Manometer *C*	Manometer *B*
1.872	1.863	1.857	1.530	1.531	1.513
1.657	1.658	1.654	1.382	1.387	1.375
1.465	1.461	1.451	1.216	1.214	1.215
1.285	1.278	1.274	1.051	1.053	1.045
1.071	1.076	1.071	0.948	0.956	0.949
0.877	0.878	0.877	0.836	0.837	0.830
0.768	0.773	0.770	0.727	0.735	0.730
0.665	0.669	0.667	0.503	0.508	0.502
0.560	0.570	0.568	0.397	0.400	0.397
0.462	0.468	0.460	0.286	0.290	0.284
0.368	0.369	0.367	0.224	0.225	0.220
0.175	0.170	0.168	0.103	0.109	0.105

Sp. Gr. Gasolene = 0.735 Sp. Gr. Gasolene = 0.736

vapor tension of either vapor in the manometer had no appreciable effect on the accuracy of the readings.

29 The large bore of the glass tubes (about ½ in. inside diameter) reduced any effect of capillary attraction to a negligible quantity and provided a reservoir of air which made the gages less sensitive to minor variations in the velocity head and therefore facilitated accurate reading.

DESCRIPTIONS OF THE EXPERIMENTS

30 The general plan of the experiments was to calibrate each pitot tube against the Thomas meter under approximately similar conditions. Two series of eight or nine tests each were made with each tube, one series being made with the end of the pipe open (full gate), which provided the condition of high velocity with low static pressure; and the other series being made with the opening in the end of the pipe restricted to one-half the pipe

area (half gate), thus providing the condition of low velocity and high static pressure. During each test the speed of the fan was kept constant for the length of time necessary to obtain all readings (20 to 30 minutes).

31 The procedure was as follows: The Thomas meter was "balanced" by causing air to flow through the pipe when there was no current flowing through the heater and the resistance R_t (Fig. 5) was out of circuit; and by moving the adjustable clamp contact until the galvanometer came to a balance. Several half days were consumed in checking and rechecking the first balance point, but ordinarily 30 minutes preceding and following a series of tests was sufficient to show that it had not changed. At frequent intervals a half day's run was made to check the original balance point. After the Thomas meter was once balanced it was found that it was not necessary to move the adjustable clamp contact again, showing that the electrical apparatus was not affected appreciably by a change in temperature from 60 deg. to 100 deg. fahr.

32 Manometers B and C, Fig. 1 and 13, were filled with gasolene of known specific gravity and carefully adjusted until their readings agreed on the average with the differential hook gage, Fig. 14. The pitot tube was tested for leaks, placed in position in the pipe and properly connected to the manometers by small rubber tubing, after which this rubber tubing was tested for air leaks.

33 When all was in readiness the fan was brought to the speed desired for the first test, the resistance R_t (Fig. 5) was placed in the thermometer circuit of the Thomas meter, the switch to the heater circuit was closed and the electrical energy to the heater regulated by the water rheostat until the galvanometer again came to a balance. Then keeping the speed constant and the galvanometer balanced, the pitot tube was placed successively at the proper points across the two diameters of the pipe and readings of the velocity head were obtained at each point. During the intervals when the manometers were coming to rest all necessary readings of pressure, temperature, revolutions per minute of fan, taken by a hand revolution counter, and volts and amperes in the heater circuit of the Thomas meter were obtained. The fact that the direct current used was furnished by a turbo-generator set having excellent voltage regulation and which supplied current for no other machines, contributed greatly

to the constancy of conditions and to the ease of obtaining accurate data.

34 When all data had been obtained for the first test the speed of the fan was increased until the tachometer needle pointed to the next determined speed, the current to the Thomas

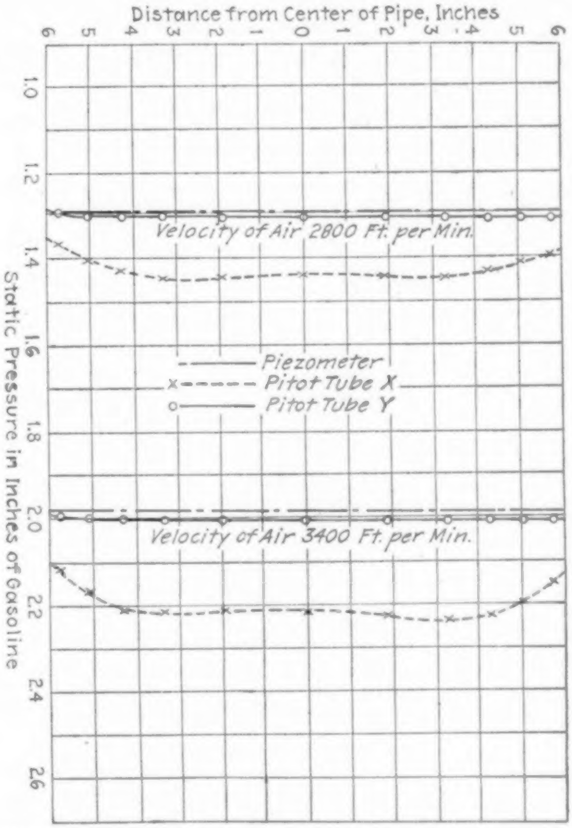


FIG. 15 STATIC TRAVERSES OF PIPE TAKEN BY PITOT TUBES X AND Y UNDER THE SAME CONDITIONS OF VELOCITY AND STATIC PRESSURE

meter was regulated until the galvanometer balanced and then all readings were obtained as before.

35 At the end of a day's run the resistance R_t (Fig. 5) was cut out of circuit and the Thomas meter balanced with no current flowing through the heater, the manometers B and C were calibrated by means of the differential hook gage and the gasolene emptied out of the manometers to check its specific gravity.

LIMITS OF ACCURACY

36 The greatest possible error in the Thomas meter was the personal error of reading the volts and amperes to the heater and this error is estimated to be under 4/10 of 1 per cent. The thermometers used were carefully selected and calibrated by means of a standard thermometer from the Bureau of Standards, and were known to be accurate for the temperatures read during these experiments. The static and barometric pressures were readable to within 1/20 of 1 per cent.

37 From an inspection of the two calibrations given in Table

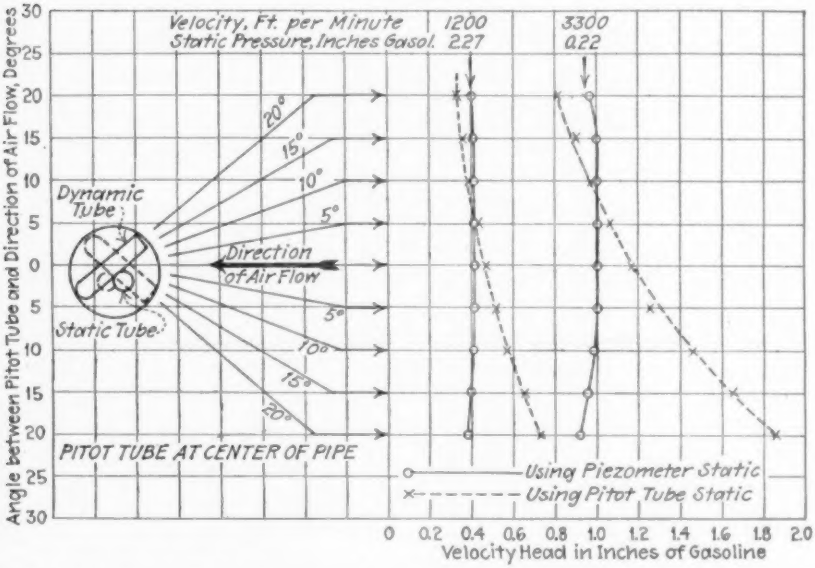


Fig. 16 Pitot Tube *L*. Diagram showing the Effect of Varying the Angle between the Pitot Tube and the Direction of Air Flow

1, of the manometers *B* and *C* used to find the pitot tube velocity heads, it will be seen that the manometer readings may be in error on the average as much as 1 per cent. Since the velocity of air flowing through the pipe varies as the square root of the velocity head, an error of 1 per cent in the velocity head means an error of ½ of 1 per cent in the velocity itself.

38 The accuracy of the measuring instruments used is much greater than can be expected from the pitot tube as a means of measuring gases, due to the uncontrollable variation in the air

flow. The considerations which may prevent even an absolutely correct pitot tube from giving true results are as follows:

a The air flows through the pipe in a wave or spiral motion and at no time is the velocity uniformly distributed across the pipe, being greater in one quarter than in the other three quarters; the quarter of highest velocities may or may not be on the diameters where pitot tube readings are being taken.

b The velocities on the diameters where pitot tube readings are being taken may be constantly varying during the period of time necessary to obtain the readings; thus the average of all readings may be slightly too large or too small.

c The air flow can only approach, never reach, the ideal conditions of parallel flow, and the pitot tube is correct, theoretically, only when the tube is exactly parallel to the current of air.

39 From these considerations, together with the author's experience in this line of work, it is estimated that the results obtained in measuring gases by an absolutely correct pitot tube may vary 1 per cent, more or less, from the correct results. Of course, the average of a large number of tests should be more nearly correct than this, for the plus errors will probably neutralize the minus errors.

40 The Thomas meter is not affected by any of these irregularities in air flow because the resistance thermometers in the form of screens across the gas passage automatically integrate the varying temperature differences resulting from heating a non-uniform current of gas or air (see Appendix No. 1).

CALCULATIONS

41 Not only was every precaution taken to obtain correct data, but the calculations were made with the same degree of refinement. Since the obtaining of accurate results by both the Thomas meter and the pitot tube depended upon the use of correct values of the properties of air, a thorough study was made of these. As a result of this study the author devised and constructed the large chart shown in Appendix No. 3, values taken from which agree almost exactly with the tables published by the Department of the Navy.[1] This chart is not only an exceedingly

[1] Dept. of Navy, Bureau of Construction and Repair, General Specifications, Appendix 8: Instructions for Calculating and Testing Ventilation System, 1908.

valuable aid in making calculations involving the properties of air, but it also presents these properties in what is thought to be a new graphical form. The basis of the calculations and the sources of information are given in the chart itself. The detailed calculations of the results of the experiments on pitot tubes are given in Appendix No. 2.

RESULTS

42 The results of the experiments on pitot tubes are presented in Tables 2 to 15 inclusive and graphically in Figs. 17 to 25 inclusive. In order that the tabulated results may be thoroughly understood a brief explanation is necessary.

43 The test number, Column 26, is given for purposes of identification and is further explained in Appendix No. 2. The cubic feet of air per minute by all three methods, i.e., by the Thomas meter, by the pitot tube alone and by the pitot dynamic tube together with the piezometer, are given in columns 28, 29, 30, and are all reduced to the conditions of temperature, pressure and humidity as determined at the section where the pitot tube is inserted.

44 Columns 31 to 35 inclusive are explained by their use in the following formulae:

$$C_1 = MC_2 \text{ or } M = \frac{C_1}{C_2}$$

$$C_1 = NC_3 \text{ or } N = \frac{C_1}{C_3}$$

$$C_3 = QC_2 \text{ or } Q = \frac{C_3}{C_2}$$

$$V_1 = \sqrt{2gh_1} = \sqrt{2gUh_3} \text{ or } U = \frac{h_1}{h_3}$$

$$V_2 = \sqrt{2gh_2} = \sqrt{2gZh_4} \text{ or } Z = \frac{h_2}{h_4}$$

where

C_1 = cu. ft. of air per minute by Thomas meter

C_2 = cu. ft. of air per minute by the pitot tube using the pitot static pressure

C_2 = cu. ft, of air per minute by the pitot tube using the

C_3 = cu. ft. of air per minute by pitot dynamic tube and the piezometer static pressure

V_1 and V_2 = velocity of the air flowing in ft. per sec.

h_1 = mean velocity head in ft. of air flowing, obtained by the pitot tube using the pitot static pressure

h_3 = velocity head in ft. of air at the center of the pipe obtained in the same manner as h_1

h_2 = mean velocity head in ft. of air flowing, obtained by the pitot dynamic tube and the piezometer static pressure

h_4 = velocity head in ft. of air at the center of the pipe, obtained in the same manner as h_2

45 *M* and *N* are therefore coefficients by which the actual results obtained by the pitot tube must be multiplied to obtain the correct flow of gas. *Q* gives the coefficient by which the results from the pitot tube alone must be multiplied to obtain the same discharge as that shown by the pitot dynamic tube and the piezometer. *U* and *Z* state the relations between the velocity head at the center of the pipe and the mean velocity head as determined by the two methods used in the experiments.

46 From a study of the summary, Table 2, the following general results and conclusions may be stated:

a The pitot tube as a means of measuring gases is reliable within approximately 1 per cent when the static pressure is correctly obtained and when all readings are taken with a sufficient degree of refinement; in order to obtain this degree of accuracy the pitot tube should be preceded by a length of pipe 20 to 38 times the pipe diameter in order to make the flow of gas as nearly uniform across the section of the pipe as possible.

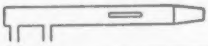
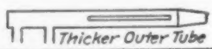
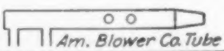
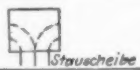
b All the methods of obtaining the dynamic head used in these experiments, including the Stauscheibe, give accurate results.

c The most reliable and accurate means of obtaining the static pressure is the piezometer or its equivalent, the results of 138 separate tests using the piezometer static pressure agreeing with the Thomas meter within an average of 0.33 per cent; these results show beyond any doubt that the static pressure is constant across any section of a pipe in which gas is flowing at a uniform rate.

d Of the methods of obtaining the static pressure by the pitot tube itself, the most reliable and accurate is by means of a very small hole in a perfectly smooth surface, as in pitot tube *Y*.

e The long slots for obtaining the static pressure are

TABLE 2 SUMMARY

	Name of Tube	*M*	*N*	*Q*	*U*	*Z*
	*A*	1.0614	1.0175	1.0413	0.7696	0.7936
	*B*	......	......	1.0576	0.8019	0.7898
	*C*	......	......	1.0343	0.7996	0.7880
	**C*	1.0346	0.9952	1.0384	0.8107	0.8066
	*D*	......	......	1.0369	0.7927	0.7942
	*E*	......	......	1.0489	0.7885	0.7898
	*X*	1.0987	0.9925	1.1074	0.7696	0.7867
	*G*	1.0076	1.0085	0.9989	0.8100	0.8164
	*H*	1.0218	1.0152	1.0065	0.7966	0.7957
	*Y*	1.0024	1.0017	0.9992	0.8002	0.7996
	*K*	1.0669	0.9924	1.0626	0.7617	0.8115
	*L*	1.0104	1.0032	1.0064	0.7593	0.8112
Average		......	1.0033	......	0.7884	0.7986
	*S*	0.9861	1.0016	0.9844	0.7780	0.8228

not reliable and give results which are in error from 3.5 to 10 per cent. The fact that slots do not give correct results is further illustrated by Fig. 15. The length of the slots or the thickness of the outer tube do not appear to affect the accuracy of the tube.

f The beveled tube for obtaining the static pressure as used in pitot tubes *K* and *L* is not reliable. A very slight change in the angle of bevel produces an appreciable change in the result. In taking a traverse of a pipe the sides of the pipe affect the readings. But the greatest error is produced by the uncertainty as to whether the tube is pointing directly upstream. The effect of allowing the tube to point at an angle with the direction of flow is shown by Fig. 16, where it is seen that if the tube is off 20 deg. in one direction an error of 85 per cent in the velocity head is introduced.

g The Stauscheibe gives accurate results using either the static reading from the Stauscheibe and the special formula (Par. 79): or by using the piezometer static with the usual formula for the pitot tubes. In the first case the agreement is within 1.4 per cent and in the second within 0.16 per cent, as shown in Table 15 and Fig. 25.

h It appears that an approximate relation exists between the mean velocity head of a gas flowing through the pipe and the velocity head found by placing the tube at the center of the pipe. For a 12-in. galvanized iron pipe results within 2 per cent may be expected from using the formula

$$v = \sqrt{(2g)\ (0.80)\ h_c}$$

where

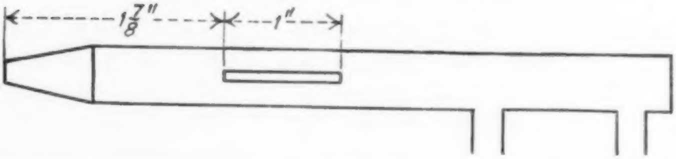
v = velocity in ft. per second

g = 32.2 ft. per second per second

h_c = velocity head in in. of gasolene at the center of the pipe obtained in a correct manner

47 The writer wishes to acknowledge the valuable help and suggestions given by Professors Carl. C. Thomas and A. G. Christie of the University of Wisconsin, and by Mr. J. C. Wilson, whose untiring interest made possible the success of these experiments. He also wishes to acknowledge his indebtedness to Mr. F. Lorig, who aided in obtaining much of the data, and to others who loaned or contributed apparatus or suggestions.

TABLE 3 RESULTS OF PITOT TUBE *A*



Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
A-1-F	795	1710	1655	1680	1.043	1.017	1.015	0.773	0.790
A-2-F	870	1910	1828	1870	1.055	1.021	1.023	0.748	0.788
A-3-F	947	2085	1995	2058	1.045	1.013	1.030	0.767	0.789
A-4-F	1031	2265	2160	2237	1.047	1.013	1.034	0.769	0.784
A-5-F	1118	2487	2340	2440	1.063	1.019	1.043	0.768	0.792
A-6-F	1191	2675	2513	2620	1.064	1.020	1.043	0.775	0.793
A-7-F	1273	2885	2685	2805	1.074	1.028	1.044	0.773	0.788
A-8-F	1370	3095	2880	3005	1.074	1.029	1.044	0.780	0.787
A-9-F	1458	3300	3080	3240	1.070	1.017	1.051	0.788	0.798
Average..	....	....	....	....	1.0595	1.0197	1.0363	0.7712	0.7900
A-1-½	794	1080	1036	1068	1.049	1.010	1.037	0.775	0.813
A-2-½	875	1178	1138	1171	1.035	1.005	1.030	0.772	0.793
A-3-½	951	1315	1240	1280	1.059	1.026	1.032	0.772	0.778
A-4-½	1028	1420	1371	1399	1.035	1.014	1.020	0.793	0.792
A-5-½	1107	1552	1452	1510	1.068	1.025	1.040	0.752	0.780
A-6-½	1190	1673	1573	1662	1.063	1.005	1.056	0.775	0.810
A-7-½	1266	1803	1663	1770	1.090	1.023	1.064	0.765	0.798
A-8-½	1371	1942	1782	1902	1.090	1.020	1.066	0.768	0.805
A-9-½	1447	2049	1892	2030	1.083	1.010	1.072	0.760	0.805
Average..........		....	....	....	1.0633	1.0153	1.0463	0.7680	0.7971
Net Average......		....	....	....	1.0614	1.0175	1.0413	0.7696	0.7936

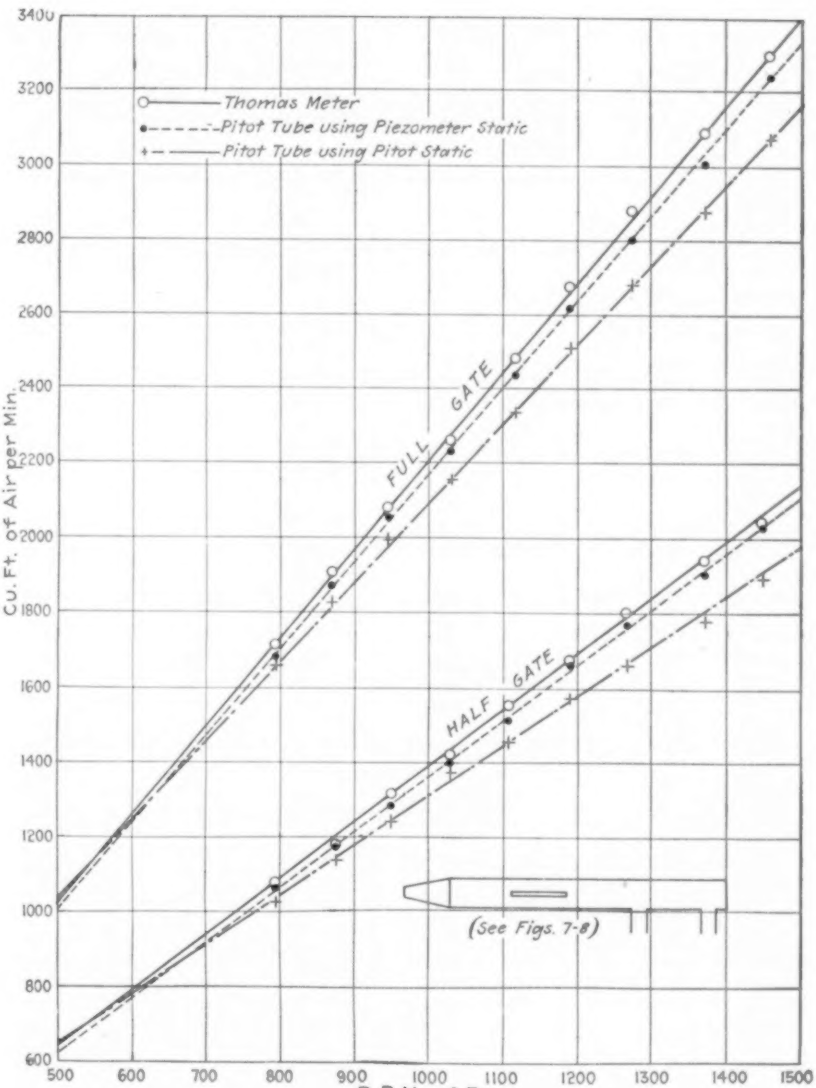
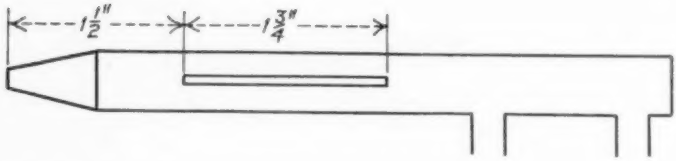


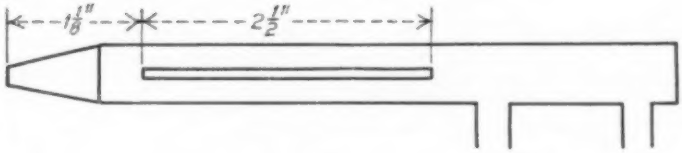
FIG. 17 PITOT TUBE *A*

TABLE 4 RESULTS OF PITOT TUBE *B*



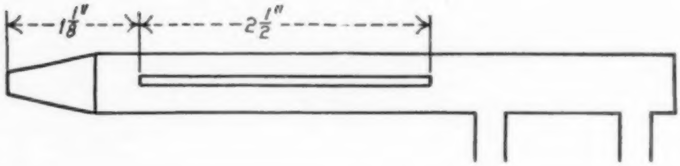
Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
B-1-F	780	....	1622	1714	.....	.....	1.056	0.802	0.792
B-2-F	877	....	1792	1889	.....	.....	1.053	0.802	0.780
B-3-F	944	....	1960	2080	.....	.....	1.061	0.798	0.788
B-4-F	1023	....	2140	2265	.....	.....	1.058	0.797	0.779
B-5-F	1110	....	2317	2470	.....	.....	1.058	0.800	0.790
B-6-F	1194	....	2488	2660	.....	.....	1.068	0.808	0.792
B-7-F	1275	....	2650	2835	.....	.....	1.069	0.812	0.787
B-8-F	1375	....	2820	3040	.....	.....	1.076	0.813	0.793
B-9-F	1458	....	3007	3240	.....	.....	1.076	0.815	0.795
Average..	....	....	....	....	.....	.....	1.0639	0.8052	0.7885
B-1-½	784	....	1020	1050	.....	.....	1.030	0.786	0.788
B-2-½	870	....	1118	1163	.....	.....	1.039	0.784	0.782
B-3-½	950	....	1217	1280	.....	.....	1.050	0.788	0.802
B-4-½	1030	....	1318	1377	.....	.....	1.045	0.792	0.778
B-5-½	1104	....	1432	1508	.....	.....	1.053	0.813	0.798
B-6-½	1185	....	1540	1628	.....	.....	1.057	0.810	0.790
B-7-½	1266	....	1648	1740	.....	.....	1.056	0.805	0.785
B-8-½	1348	....	1760	1875	.....	.....	1.064	0.810	0.798
B-9-½	1447	....	1883	2010	.....	.....	1.068	0.809	0.797
Average..........		....	....	....	.....	.....	1.0513	0.7986	0.7910
Net Average......		....	....	....	.....	.....	1.0576	0.8019	0.7898

TABLE 5 RESULTS OF PITOT TUBE C



Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
C-1-F	790	....	1623	1672	.....	.....	1.030	0.793	0.777
C-2-F	872	....	1793	1860	.....	.....	1.036	0.793	0.762
C-3-F	950	....	1975	2042	.....	.....	1.034	0.807	0.795
C-4-F	1030	....	2160	2228	.....	.....	1.032	0.800	0.782
C-5-F	1124	....	2348	2432	.....	.....	1.034	0.804	0.789
C-6-F	1198	....	2505	2600	.....	.....	1.038	0.799	0.786
C-7-F	1289	....	2685	2780	.....	.....	1.034	0.803	0.789
C-8-F	1378	....	2870	2987	.....	.....	1.041	0.806	0.793
C-9-F	1466	....	3065	3195	.....	.....	1.042	0.804	0.793
Average..	....	....	....	....	.....	.....	1.0357	0.8010	0.7847
C-1-½	793	....	1024	1044	.....	.....	1.020	0.800	0.785
C-2-½	866	....	1131	1150	.....	.....	1.016	0.796	0.781
C-3-½	947	....	1239	1268	.....	.....	1.023	0.796	0.788
C-4-½	1033	....	1340	1380	.....	.....	1.031	0.807	0.793
C-5-½	1115	....	1450	1505	.....	.....	1.038	0.800	0.794
C-6-½	1194	....	1562	1620	.....	.....	1.038	0.797	0.789
C-7-½	1271	....	1672	1737	.....	.....	1.038	0.795	0.790
C-8-½	1360	....	1800	1872	.....	.....	1.041	0.798	0.795
C-9-½	1450	....	1920	2015	.....	.....	1.050	0.802	0.806
Average...........		....	....	....	.....	.....	1.0328	0.7983	0.7912
Net Average......		....	....	....	.....	.....	1.0343	0.7996	0.7880

TABLE 6 RESULTS OF PITOT TUBE *C



Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
*C-1-F	...	....	....	....	.....	.....	.....	.....	.....
*C-2-F	850	1693	1657	1708	1.022	0.992	1.030	0.809	0.794
*C-3-F	915	1850	1818	1872	1.018	0.989	1.030	0.801	0.792
*C-4-F	994	2025	1970	2039	1.027	0.993	1.033	0.809	0.803
*C-5-F	1068	2202	2133	2210	1.032	0.996	1.034	0.808	0.800
*C-6-F	1150	2370	2297	2385	1.033	0.994	1.037	0.813	0.804
*C-7-F	1214	2540	2468	2560	1.029	0.994	1.036	0.817	0.808
*C-8-F	1290	2720	2633	2740	1.033	0.994	1.040	0.814	0.806
*C-9-F	1385	2935	2820	2955	1.040	0.994	1.027	0.815	0.812
Average........		....	....	....	1.0292	0.9932	1.0336	0.8107	0.8024
*C-1-½	...	....	....	....	.....	.....	.....	.....	.....
*C-2-½	847	1143	1105	1126	1.033	1.015	1.020	0.807	0.807
*C-3-½	914	1233	1202	1239	1.025	0.996	1.030	0.808	0.805
*C-4-½	987	1326	1290	1340	1.029	0.990	1.037	0.808	0.803
*C-5-½	1066	1440	1386	1448	1.040	0.996	1.044	0.817	0.813
*C-6-½	1145	1544	1479	1553	1.044	0.995	1.049	0.817	0.817
*C-7-½	1214	1643	1575	1658	1.043	0.992	1.052	0.808	0.808
*C-8-½	1298	1773	1675	1780	1.058	0.996	1.063	0.802	0.818
*C-9-½	1386	1885	1800	1890	1.047	0.998	1.050	0.818	0.815
Average..........		....	....	....	1.0399	0.9972	1.0431	0.8106	0.8107
Net Average......		....	....	....	1.0346	0.9952	1.0384	0.8107	0.8066

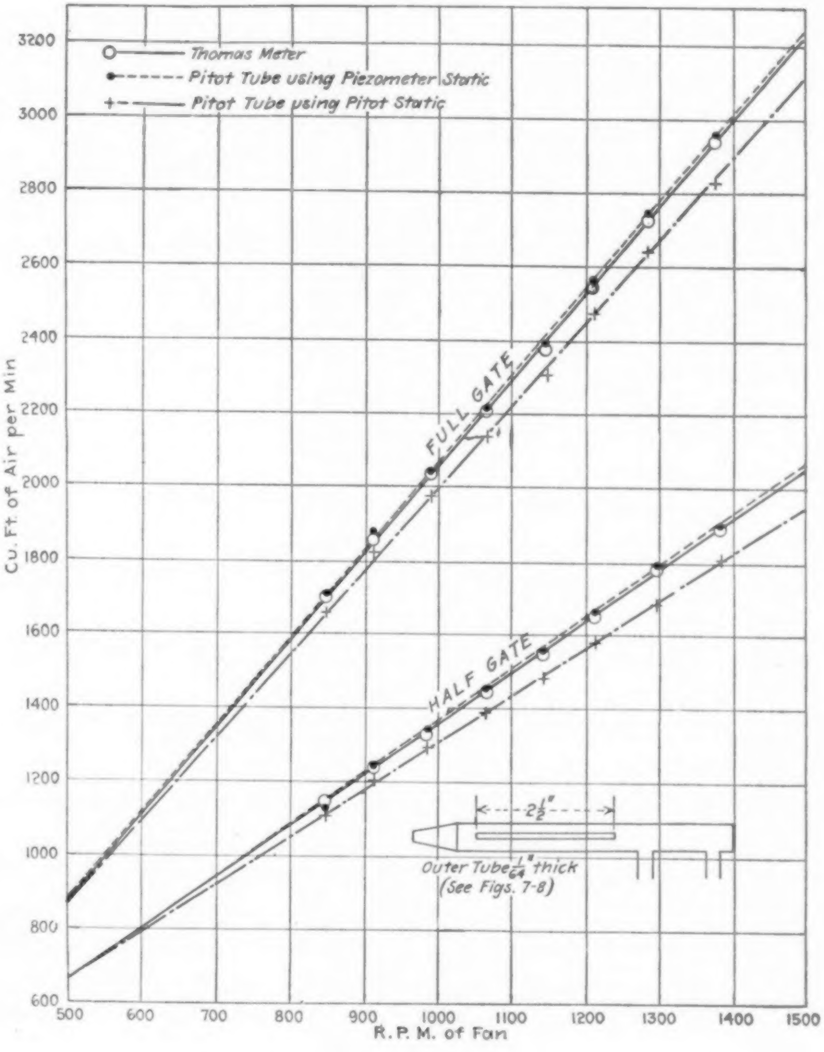
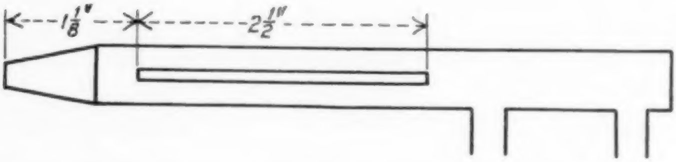


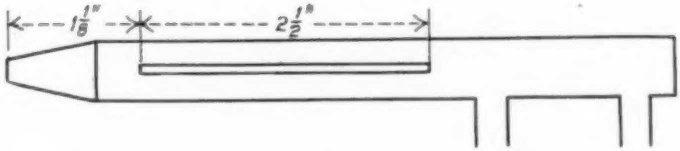
FIG. 18 PITOT TUBE* C

TABLE 7 RESULTS OF PITOT TUBE *D*



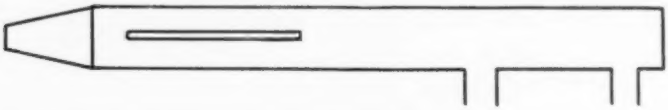
Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
D-1-F	792	....	1583	1620	.....	.....	1.024	0.779	0.754
D-2-F	873	....	1752	1805	.....	.....	1.030	0.789	0.777
D-3-F	954	....	1932	2005	.....	.....	1.037	0.790	0.792
D-4-F	1029	....	2100	2160	.....	.....	1.028	0.793	0.781
D-5-F	1118	....	2295	2367	.....	.....	1.031	0.792	0.781
D-6-F	1202	....	2470	2550	.....	.....	1.032	0.795	0.788
D-7-F	1283	....	2660	2750	.....	.....	1.033	0.796	0.790
D-8-F	1377	....	2850	2950	.....	.....	1.034	0.798	0.792
D-9-F	1465	....	3045	3150	.....	.....	1.035	0.798	0.798
Average..	....	....	....	....	.....	.....	.....	.....	.....
D-1-½	792	....	984	1019	.....	.....	1.035	0.772	0.782
D-2-½	871	....	1097	1127	.....	.....	1.034	0.787	0.790
D-3-½	955	....	1200	1254	.....	.....	1.045	0.795	0.808
D-4-½	1030	....	1303	1348	.....	.....	1.035	0.790	0.786
D-5-½	1121	....	1431	1487	.....	.....	1.039	0.812	0.807
D-6-½	1204	....	1526	1593	.....	.....	1.043	0.795	0.792
D-7-½	1290	....	1635	1710	.....	.....	1.045	0.798	0.796
D-8-½	1379	....	1760	1850	.....	.....	1.050	0.793	0.795
D-9-½	1473	....	1880	1985	.....	.....	1.054	0.796	0.796
Average..........		....	....	....	.....	.....	1.0422	0.7931	0.7947
Net Average......		....	....	....	.....	.....	1.0369	0.7927	0.7942

TABLE 8 RESULTS OF PITOT TUBE *E*



Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			*M*	*N*	*Q*	*U*	*Z*
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
E-1-F	790	....	1577	1626	.....	.....	1.031	0.774	0.779
E-2-F	872	....	1730	1797	.....	.....	1.039	0.779	0.781
E-3-F	951	....	1895	1975	.....	.....	1.042	0.783	0.786
E-4-F	1031	....	2060	2153	.....	.....	1.044	0.790	0.791
E-5-F	1116	....	2242	2355	.....	.....	1.050	0.787	0.787
E-6-F	1202	....	2418	2543	.....	.....	1.053	0.788	0.788
E-7-F	1282	....	2592	2735	.....	.....	1.054	0.788	0.790
E-8-F	1380	....	2850	2960	.....	.....	1.039	0.792	0.793
E-9-F	1465	....	3050	3155	.....	.....	1.033	0.791	0.795
Average	....	....	....	....	.....	.....	1.0429	0.7859	0.7879
E-1-½	792	....	1008	1057	.....	.....	1.050	0.790	0.805
E-2-½	873	....	1130	1176	.....	.....	1.040	0.788	0.802
E-3-½	952	....	1224	1276	.....	.....	1.042	0.785	0.799
E-4-½	1030	....	1324	1385	.....	.....	1.046	0.785	0.795
E-5-½	1117	....	1440	1515	.....	.....	1.052	0.788	0.803
E-6-½	1201	....	1543	1630	.....	.....	1.056	0.790	0.798
E-7-½	1284	....	1655	1760	.....	.....	1.063	0.790	0.797
E-8-½	1378	....	1777	1895	.....	.....	1.065	0.800	0.808
E-9-½	1468	....	1890	2040	.....	.....	1.079	0.803	0.818
Average...........		....	....	....	.....	.....	1.0549	0.7910	0.7917
Net Average......		....	....	....	.....	.....	1.0489	0.7885	0.7898

TABLE 9 RESULTS OF PITOT TUBE *X*



Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
X-1-F	807	1605	1510	1650	1.062	0.973	1.092	0.798	0.790
X-2-F	888	1800	1655	1825	1.087	0.987	1.102	0.794	0.793
X-3-F	963	1962	1805	2000	1.087	0.982	1.108	0.793	0.795
X-4-F	1046	2140	1975	2180	1.083	0.983	1.103	0.793	0.788
X-5-F	1135	2345	2125	2380	1.102	0.985	1.120	0.770	0.793
X-6-F	1229	2545	2280	2562	1.115	0.993	1.123	0.775	0.792
X-7-F	1311	2745	2455	2740	1.117	1.002	1.116	0.772	0.779
X-8-F	1410	2940	2625	2970	1.120	0.991	1.130	0.774	0.800
X-9-F	1496	3145	2785	3165	1.128	0.993	1.135	0.777	0.798
Average	....	....	....	....	1.1001	0.9877	1.1143	0.7830	0.7920
X-1-½	785	1047	958	1037	1.093	1.010	1.083	0.738	0.767
X-2-½	876	1161	1060	1153	1.095	1.007	1.088	0.751	0.788
X-3-½	954	1270	1156	1260	1.098	1.007	1.090	0.757	0.790
X-4-½	1028	1378	1256	1380	1.098	0.999	1.098	0.755	0.780
X-5-½	1120	1490	1363	1488	1.094	1.001	1.093	0.755	0.770
X-6-½	1195	1595	1457	1609	1.095	0.992	1.103	0.758	0.790
X-7-½	1288	1713	1557	1723	1.100	0.994	1.105	0.760	0.782
X-8-½	1370	1834	1670	1870	1.099	0.983	1.120	0.765	0.772
X-9-½	1460	1960	1775	1996	1.103	0.983	1.124	0.767	0.793
Average..........		....	....	....	1.0972	0.9973	1.1004	0.7562	0.7813
Net Average......		....	....	....	1.0987	0.9925	1.1074	0.7696	0.7867

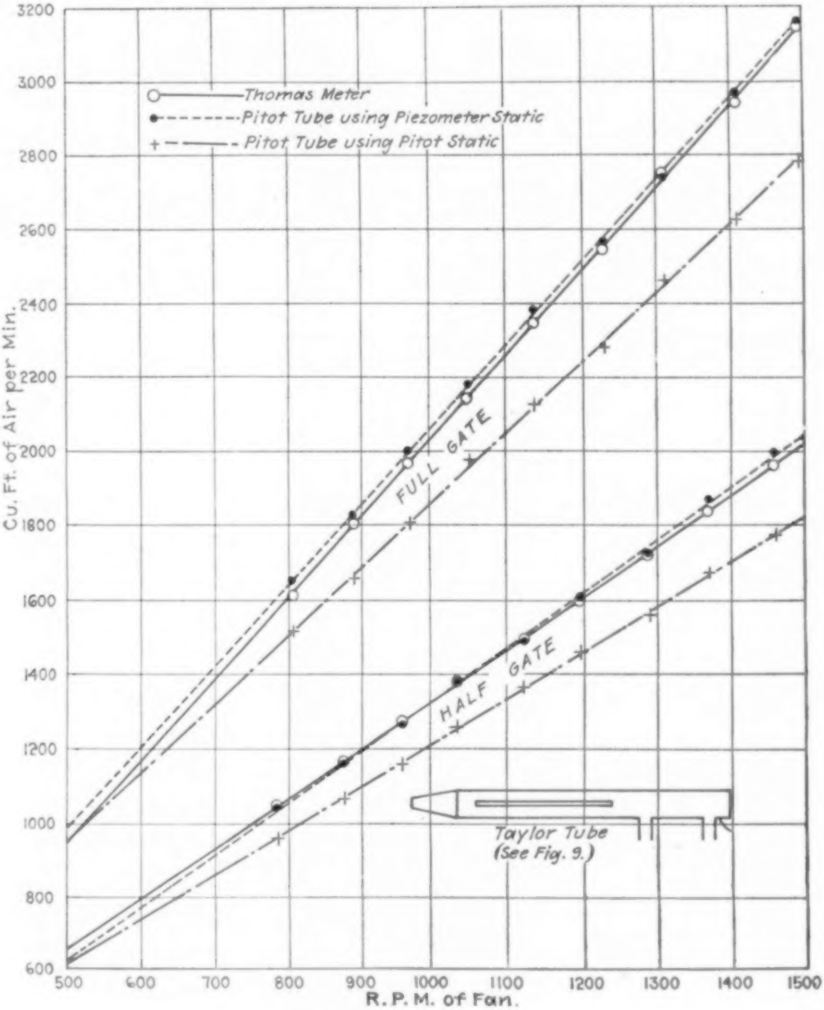
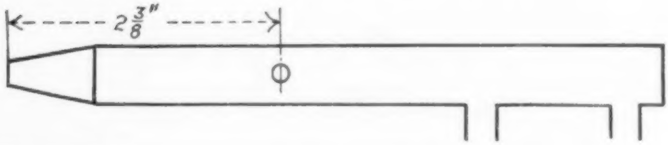


Fig. 19 Pitot Tube X

TABLE 10 RESULTS OF PITOT TUBE *G*



Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min. Thomas Meter C_1	Pitot Tube Using Pitot Static C_2	Pitot Tube Using Piezometer Static C_3	*M* $\frac{C_1}{C_2}$	*N* $\frac{C_1}{C_3}$	*Q* $\frac{C_3}{C_2}$	*U* $\frac{\text{Col. 18}}{\text{Col. 20}}$	*Z* $\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
G–1–F	760	1590	1556	1550	1.021	1.026	0.997	0.798	0.806
G–2–F	835	1764	1722	1716	1.023	1.027	0.997	0.792	0.797
G–3–F	912	1912	1882	1870	1.021	1.022	0.994	0.811	0.812
G–4–F	984	2060	2032	2027	1.013	1.014	0.998	0.803	0.793
G–5–F	1060	2220	2220	2215	1.000	1.002	0.998	0.802	0.797
G–6–F	1128	2406	2370	2370	1.014	1.014	1.000	0.802	0.812
G–7–F	1202	2543	2537	2535	1.003	1.003	0.999	0.811	0.817
G–8–F	1278	2718	2718	2715	1.000	1.000	0.999	0.813	0.822
G–9–F	1356	2910	2895	2900	1.004	1.003	1.002	0.812	0.822
Average........		....	....	....	1.0110	1.0123	0.9982	0.8049	0.8097
G–1–½	....	....	....	....	.....	.....	.....	.....	.....
G–2–½	836	1124	1135	1114	0.992	1.009	0.982	0.811	0.812
G–3–½	900	1217	1240	1227	0.982	0.993	0.991	0.805	0.814
G–4–½	988	1343	1331	1323	1.007	1.013	0.993	0.812	0.818
G–5–½	1060	1439	1435	1430	1.002	1.006	0.998	0.809	0.813
G–6–½	1135	1554	1538	1544	1.010	1.006	1.003	0.813	0.823
G–7–½	1210	1663	1632	1640	1.017	1.013	1.004	0.811	0.821
G–8–½	1288	1769	1753	1772	1.009	0.998	1.010	0.822	0.834
G–9–½	1370	1883	1870	1885	1.005	0.999	1.006	0.837	0.849
Average...........		....	....	....	1.0042	1.0046	0.9995	0.8150	0.8230
Net Average......		....	....	....	1.0076	1.0085	0.9989	0.8100	0.8164

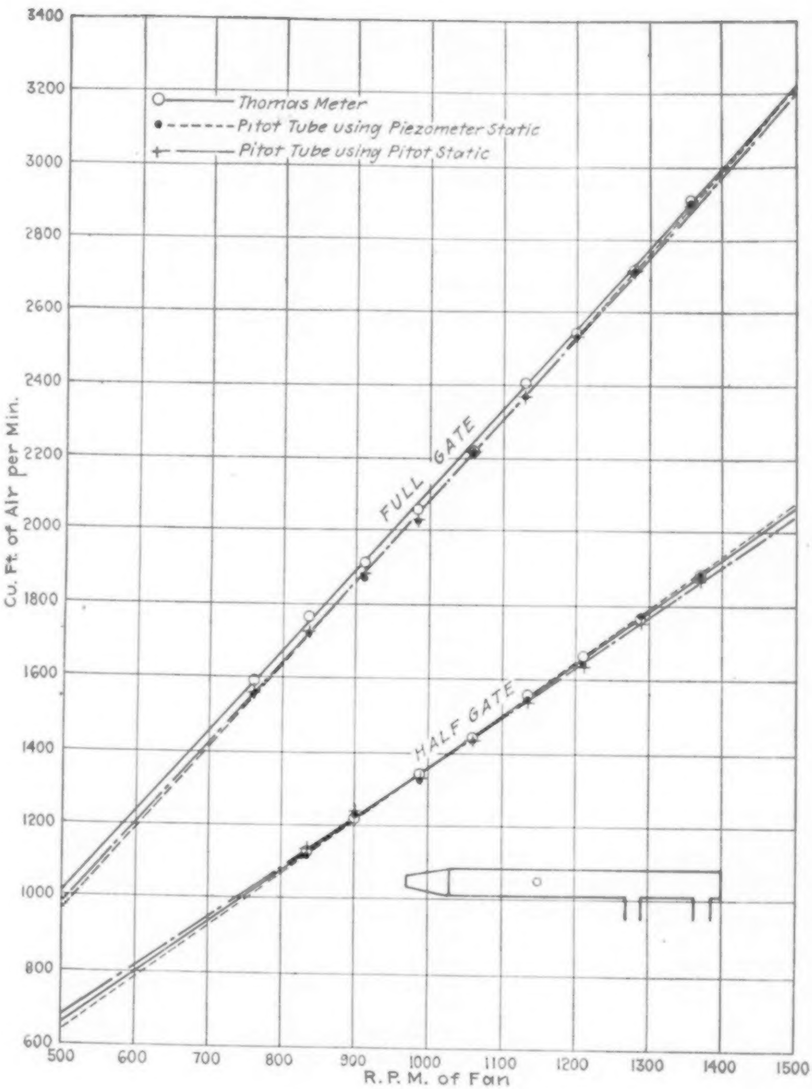
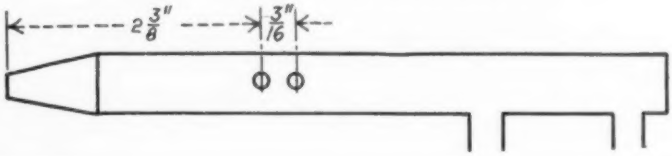


FIG. 20 PITOT TUBE G

TABLE 11 RESULTS OF PITOT TUBE *H*



Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
H-1-F	....	....	....	....	.....	.....	.....	.....	.....
H-2-F	872	1930	1922	1912	1.003	1.008	0.998	0.803	0.798
H-3-F	947	2108	2123	2123	0.993	0.993	1.000	0.830	0.826
H-4-F	1024	2300	2275	2269	1.010	1.013	0.998	0.796	0.790
H-5-F	1110	2508	2465	2463	1.017	1.018	0.999	0.795	0.792
H-6-F	1195	2710	2656	2653	1.016	1.017	0.999	0.792	0.790
H-7-F	1273	2910	2855	2855	1.019	1.019	1.000	0.799	0.798
H-8-F	1375	3120	3062	3060	1.018	1.019	0.999	0.800	0.800
H-9-F	1456	3330	3265	3270	1.019	1.018	1.001	0.798	0.803
Average..	....	....	....	....	1.0120	1.0131	0.9992	0.8016	0.7996
H-1-½	....	....	....	....	.....	.....	.....	.....	....
H-2-½	870	1203	1170	1176	1.026	1.023	1.006	0.788	0.793
H-3-½	948	1313	1274	1284	1.028	1.021	1.007	0.775	0.780
H-4-½	1029	1425	1390	1403	1.025	1.015	1.010	0.787	0.787
H-5-½	1110	1570	1523	1535	1.030	1.022	1.007	0.802	0.791
H-6-½	1185	1685	1623	1654	1.036	1.018	1.020	0.797	0.795
H-7-½	1265	1793	1738	1763	1.032	1.016	1.013	0.796	0.791
H-8-½	1360	1932	1870	1910	1.033	1.010	1.021	0.798	0.801
H-9-½	1454	2072	1986	2040	1.043	1.013	1.026	0.789	0.796
Average...........		....	....	....	1.0316	1.0172	1.0137	0.7915	0.7917
Net Average......		....	....	....	1.0218	1.0152	1.0065	0.7966	0.7957

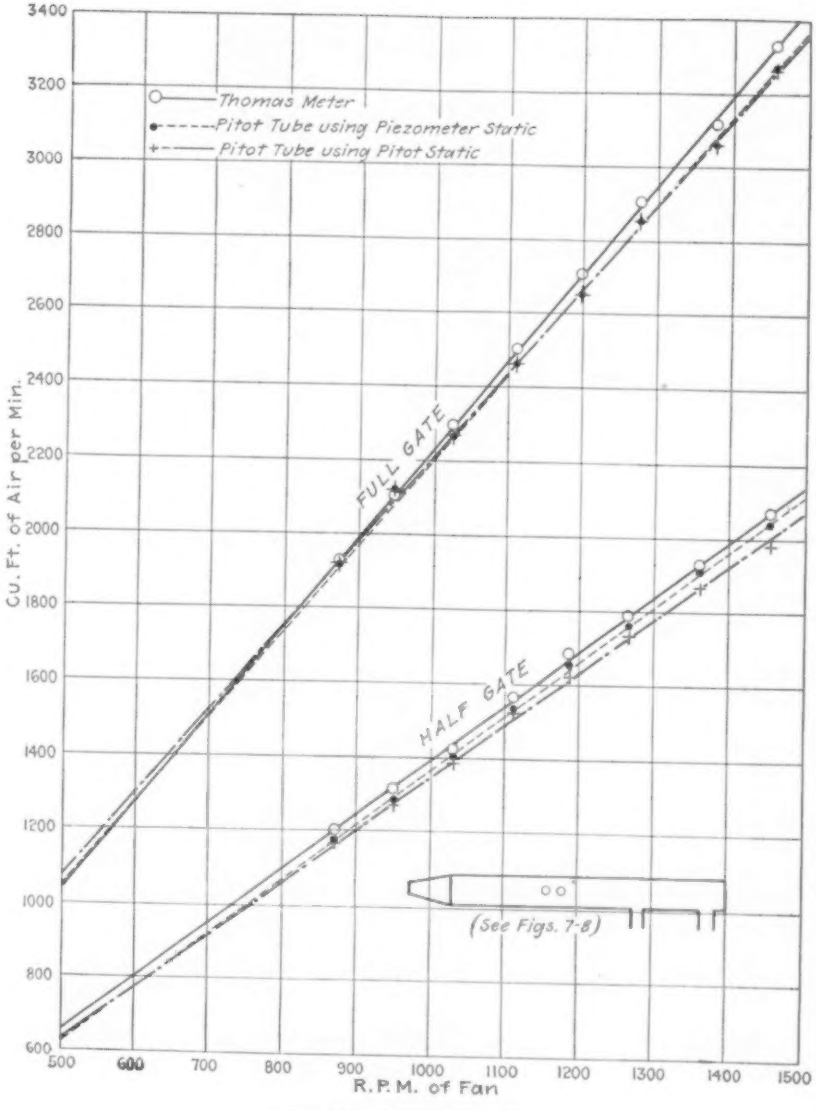
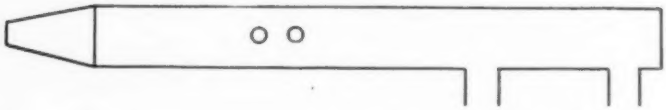


FIG. 21 PITOT TUBE *H*

TABLE 12 RESULTS OF PITOT TUBE *Y*



Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
Y-1-F	793	1635	1652	1650	0.991	0.992	0.999	0.800	0.813
Y-2-F	887	1865	1860	1860	1.002	1.002	1.000	0.799	0.808
Y-3-F	968	2033	2032	2030	1.000	1.001	0.999	0.803	0.796
Y-4-F	1050	2215	2215	2205	1.000	1.003	0.997	0.797	0.812
Y-5-F	1144	2424	2425	2422	1.000	1.001	0.999	0.801	0.795
Y-6-F	1226	2606	2608	2595	0.999	1.003	0.996	0.805	0.797
Y-7-F	1316	2800	2800	2790	1.000	1.003	0.998	0.806	0.802
Y-8-F	1409	3035	3010	2990	1.007	1.014	0.993	0.808	0.797
Y-9-F	1490	3220	3215	3195	1.001	1.007	0.994	0.806	0.798
Average..	....	....	....	....	1.000	1.0029	0.9951	0.8027	0.8020
Y-1-½	786	1025	1040	1040	0.986	0.986	1.000	0.791	0.794
Y-2-½	876	1160	1152	1152	1.007	1.007	1.000	0.797	0.790
Y-3-½	952	1271	1267	1267	1.004	1.003	0.999	0.800	0.795
Y-4-½	1026	1382	1380	1381	1.001	1.000	1.000	0.798	0.795
Y-5-½	1113	1510	1503	1508	1.004	1.001	1.003	0.802	0.802
Y-6-½	1195	1636	1620	1630	1.009	1.003	1.004	0.803	0.802
Y-7-½	1287	1756	1740	1750	1.009	1.003	1.004	0.798	0.796
Y-8-½	1370	1895	1872	1886	1.011	1.004	1.006	0.799	0.799
Y-9-½	1462	2015	1990	2020	1.012	0.998	0.014	0.802	0.802
Average..........		....	....	....	1.0048	1.0005	1.0033	0.7990	0.7972
Net Average......		....	....	....	1.0024	1.0017	0.9992	0.8002	0.7996

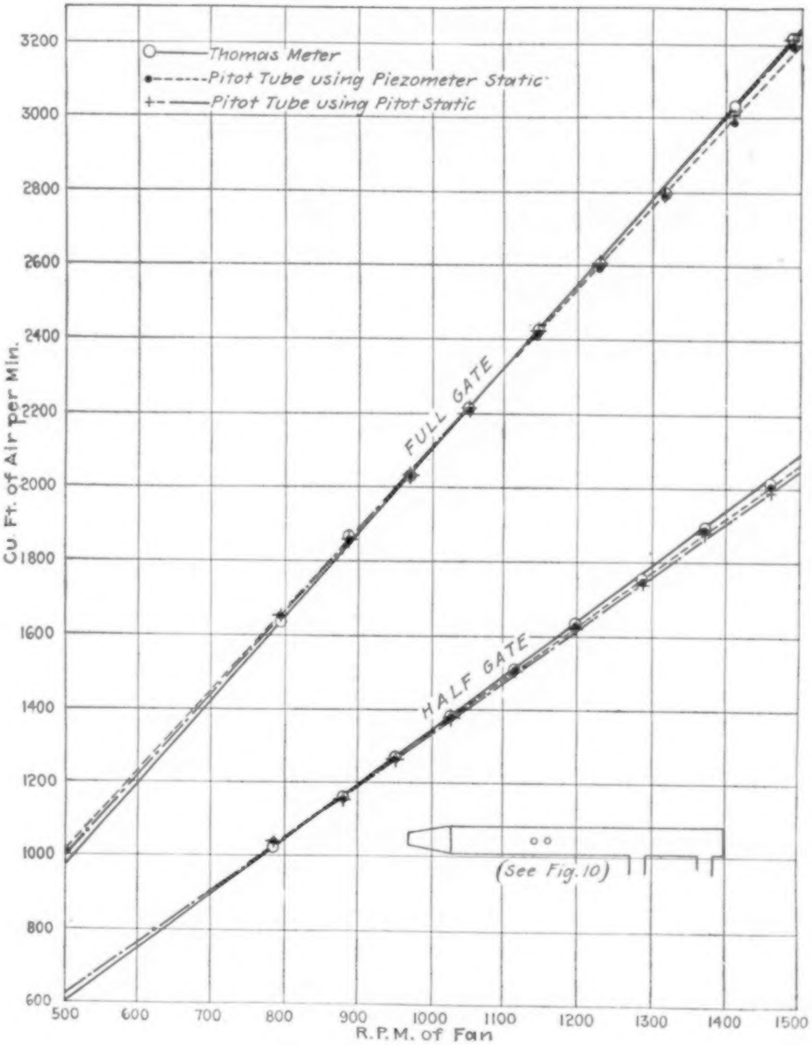


Fig. 22 Pitot Tube Y

TABLE 13 RESULTS OF PITOT TUBE *K*

Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
K-1-F	....	....	....	....	.....	.....	.....	.....	.....
K-2-F	840	1707	1542	1673	1.107	1.020	1.085	0.773	0.808
K-3-F	910	1863	1675	1826	1.112	1.019	1.090	0.767	0.810
K-4-F	985	2020	1832	1985	1.102	1.017	1.083	0.771	0.813
K-5-F	1060	2186	2003	2160	1.091	1.012	1.077	0.763	0.818
K-6-F	1140	2363	2162	2308	1.091	1.022	1.067	0.788	0.822
K-7-F	1215	2527	2310	2480	1.093	1.018	1.073	0.762	0.802
K-8-F	1295	2695	2487	2672	1.083	1.009	1.073	0.762	0.818
K-9-F	1375	2850	2660	2842	1.070	1.003	1.068	0.763	0.797
Average..	....	....	....	....	1.0936	1.0150	1.0770	0.7686	0.8110
K-1-½	....	....	....	....	.....	.....	.....	.....	.....
K-2-½	828	1077	1063	1098	1.015	0.983	1.033	0.758	0.808
K-3-½	900	1192	1147	1193	1.039	1.000	1.038	0.745	0.805
K-4-½	982	1307	1240	1294	1.054	1.010	1.043	0.752	0.814
K-5-½	1070	1405	1338	1413	1.050	0.995	1.054	0.758	0.815
K-6-½	1140	1498	1441	1510	1.040	0.993	1.047	0.752	0.806
K-7-½	1216	1581	1518	1595	1.042	0.992	1.050	0.752	0.812
K-8-½	1295	1675	1620	1728	1.033	0.971	1.067	0.760	0.819
K-9-½	1377	1802	1720	1812	1.048	0.995	1.053	0.761	0.817
Average...........		....	....	....	1.0401	0.9924	1.0481	0.7548	0.8120
Net Average......		....	....	....	1.[illegible]	1.0037	1.0626	0.7617	0.8115

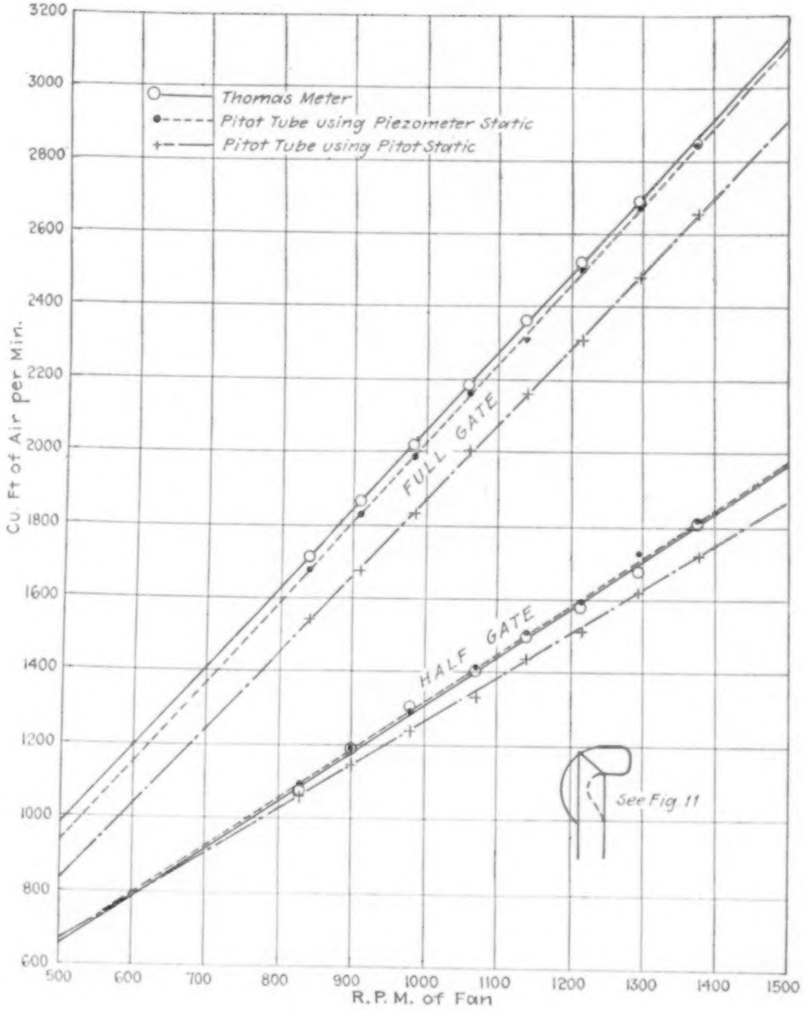


FIG. 23 PITOT TUBE *K*

TABLE 14 RESULTS OF PITOT TUBE *L*

Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	Q	U	Z
		Thomas Meter C_1	Pitot Tube						
			Using Pitot Static C_2	Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
L-1-F	....	....	....	....	.....	.....	.....	.....	.....
L-2-F	838	1733	1678	1684	1.032	1.028	1.003	0.758	0.815
L-3-F	....	....	....	....	.....	.....	.....	.....	.....
L-4-F	980	2010	1980	1993	1.014	1.007	1.006	0.757	0.798
L-5-F	1060	2200	2145	2165	1.025	1.015	1.009	0.748	0.806
L-6-F	1140	2357	2318	2335	1.016	1.008	1.007	0.753	0.810
L-7-F	1210	2517	2470	2492	1.017	1.009	1.007	0.737	0.802
L-8-F	1295	2700	2672	2685	1.010	1.004	1.004	0.757	0.809
L-9-F	1365	2878	2833	2870	1.015	1.002	1.013	0.752	0.812
Average..	....	....	....	....	1.0178	1.0104	1.0068	0.7517	0.8074
L-1-½	....	....	....	....	.....	.....	.....	.....	.....
L-2-½	828	1100	1120	1105	0.982	0.996	0.987	0.775	0.805
L-3-½	900	1209	1208	1213	1.000	0.997	1.002	0.769	0.816
L-4-½	970	1296	1300	1306	0.997	0.993	1.004	0.772	0.817
L-5-½	1055	1402	1402	1418	1.000	0.989	1.010	0.762	0.807
L-6-½	1127	1522	1500	1519	1.013	1.002	1.011	0.765	0.820
L-7-½	1210	1628	1609	1625	1.013	1.002	1.010	0.772	0.822
L-8-½	1290	1721	1710	1734	1.006	0.993	1.013	0.758	0.810
L-9-½	1360	1838	1815	1845	1.012	0.997	1.016	0.761	0.822
Average..........		....	....	....	1.0030	0.996	1.0060	0.7668	0.8150
Net Average......		....	....	....	1.0104	1.0032	1.0064	0.7593	0.8112

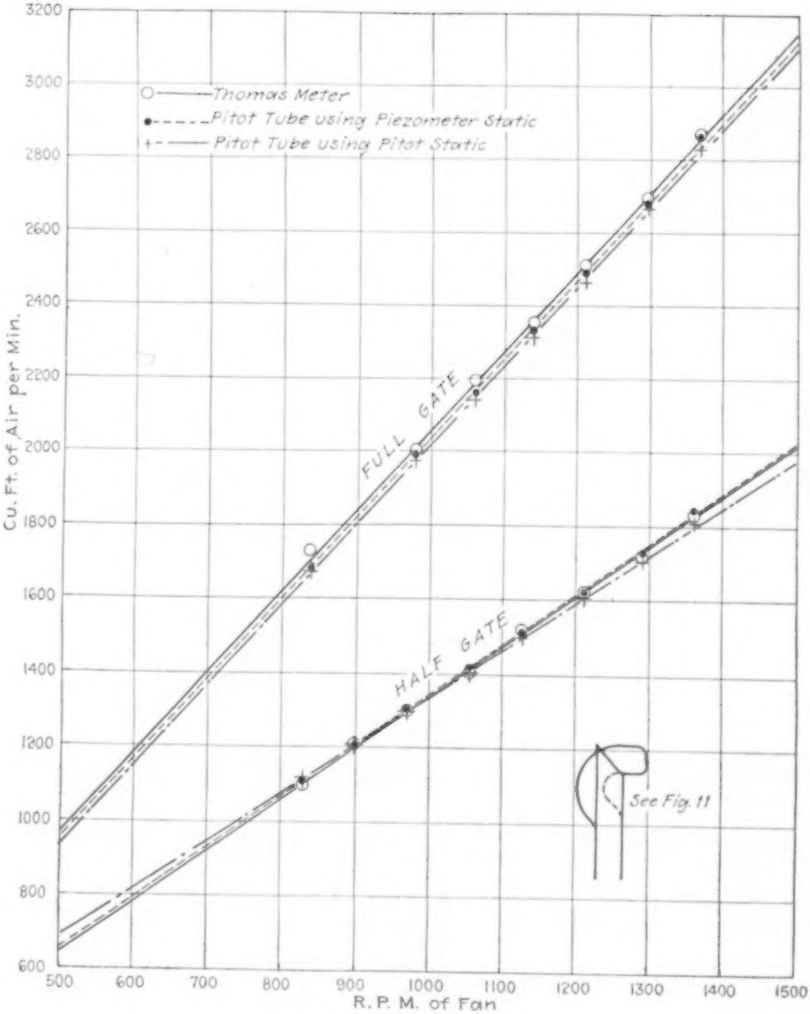
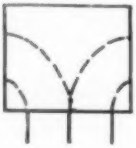


FIG. 24 PITOT TUBE L

TABLE 15 RESULTS OF STAUSCHEIBE *S*



Test No.	R.p.m. of Fan	Cu. Ft. of Air per Min.			M	N	O	U	Z
		Thomas Meter C_1	Stauscheibe: Using Stauscheibe Static Constant—0.854 C_2	Stauscheibe: Using Piezometer Static C_3	$\frac{C_1}{C_2}$	$\frac{C_1}{C_3}$	$\frac{C_3}{C_2}$	$\frac{\text{Col. 18}}{\text{Col. 20}}$	$\frac{\text{Col. 19}}{\text{Col. 21}}$
26	27	28	29	30	31	32	33	34	35
S-1-F	....	....	....	....	.....	.....	.....	.....	.....
S-2-F	841	1695	1730	1678	0.981	1.009	0.971	0.773	0.807
S-3-F	912	1858	1900	1832	0.979	1.013	0.965	0.779	0.805
S-4-F	983	2003	2060	1986	0.974	1.008	0.965	0.774	0.797
S-5-F	1070	2190	2245	2182	0.976	1.003	0.974	0.778	0.812
S-6-F	1152	2355	2420	2353	0.975	1.001	0.973	0.782	0.809
S-7-F	1220	2555	2590	2522	0.986	1.012	0.975	0.790	0.820
S-8-F	1312	2740	2800	2722	0.980	1.005	0.973	0.797	0.823
S-9-F	1400	2937	2990	2900	0.982	1.012	0.970	0.808	0.823
Average........		....	....	....	0.9791	1.0079	0.9708	0.7851	0.8245
S-1-½	....	....	....	....	.....	.....	.....	.....	.....
S-2-½	841	1093	1120	1092	0.978	1.001	0.978	0.772	0.788
S-3-½	918	1214	1223	1208	0.995	1.004	0.990	0.785	0.831
S-4-½	997	1320	1311	1320	1.005	1.000	1.005	0.767	0.840
S-5-½	1074	1417	1427	1423	0.993	0.997	0.997	0.777	0.832
S-6-½	1152	1505	1520	1520	0.992	0.991	1.000	0.772	0.824
S-7-½	1232	1629	1621	1625	1.003	1.002	1.002	0.770	0.817
S-8-½	1314	1720	1743	1750	0.988	0.985	1.004	0.768	0.815
S-9-½	1399	1843	1860	1877	0.990	0.982	1.008	0.772	0.821
Average...........		....	....	....	0.9930	0.9952	0.9980	0.7729	0.8210
Net Average......		....	....	....	0.9861	1.0016	0.9844	0.7780	0.8228

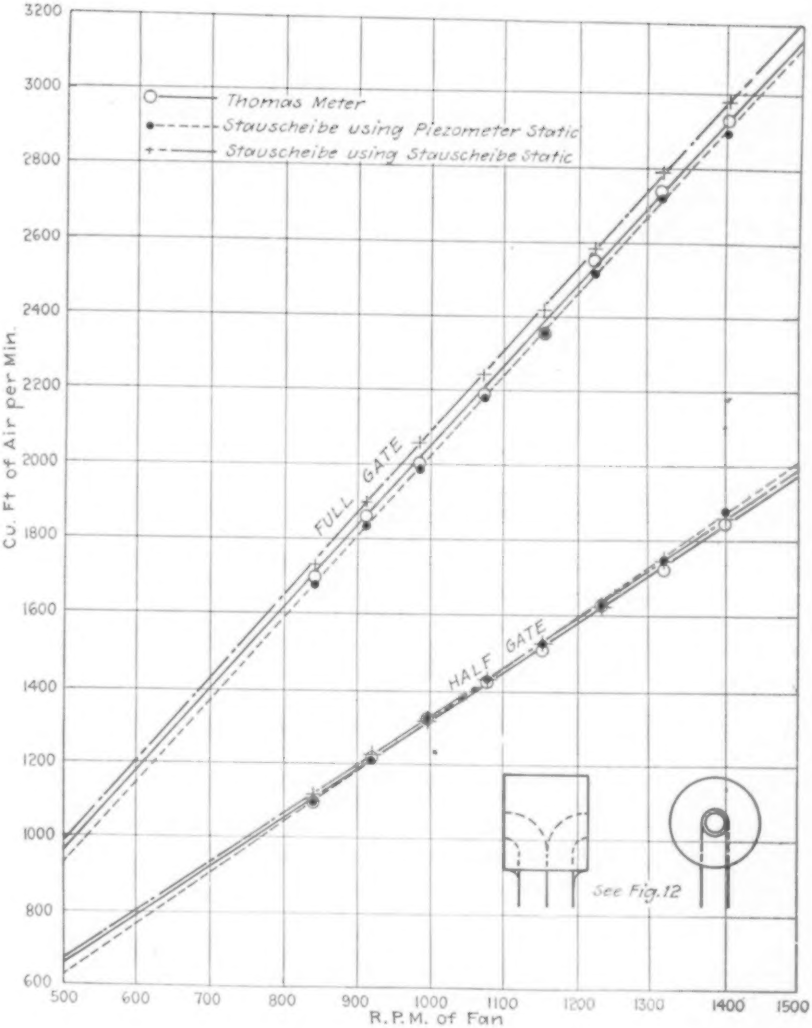


Fig. 25 Stauscheibe *S*

APPENDIX NO. 1

THE ACCURACY OF THE THOMAS METER

48 The Thomas meter has been thoroughly tested under various conditions of service and a few of the tests are briefly referred to in the following notes in order to show that it is a reliable and accurate means for measuring gases and that its use as a standard meter in these experiments with pitot tubes was justified.

TESTS AGAINST CALIBRATED PITOT TUBES [1]

49 A Thomas meter was tested by the People's Natural Gas Company of Pittsburgh at their Brave Pumping Station, in a 10-in. suction line from the gas wells to the pump and in series with a pitot tube station. This latter was located a mile and a half from the pumping station where the electric meter was installed and every precaution was taken to prevent leakage in the intervening pipe line. This pitot tube station was developed after years of experiment and at great expense. The pitot tubes in the station were carefully calibrated under working conditions and were known to give accurate results when the calibration constants were used.

50 A 45-day test was run, from April 7 to June 3, 1911, during which period the rate of flow varied from 90,000 to 640,000 cu. ft. per hour, the pressure of the gas varied from 46 to 185 lb. gage and the temperature varied from 45 to 65 deg. fahr. The results were as follows:

Total standard cu. ft. of gas by pitot tube	337,546,182
Total standard cu. ft. of gas by Thomas meter	336,732,018
Difference	814,164
Per cent of difference	0.24

A year later a test of the same meter was made without any changes of adjustment whatever and the Thomas meter was found to agree with the pitot tubes within 0.42 per cent.

TESTS AGAINST HOLDERS [1]

51 A Thomas meter is used to measure the gas in the common discharge line from the booster station of the Milwaukee Gas Light Company. The holder tests were made with the greatest care on October 13 and 14, 1911, at a time when the temperature had remained constant during the day and night for several days. Two different holders were used. One test was made at the maximum capacity of the meter when the gas was being pumped from the large holder through the meter at a pressure of about 40 in. of water. A second test was made at the minimum capacity of the

[1] Carl C. Thomas. Proc. Am. Gas Inst., vol. 7, 1912.

meter when gas was flowing from the smaller holder through the meter at holder pressure.

52 The results were as follows:

Holder used	Larger	Smaller
Duration of test	2 hr. 15 min.	5 hr.
Total cu. ft. of air by Thomas meter at 30 in. mercury and 60 deg. fahr.	1,956,000	497,500
Total cu. ft. of air by holder at 30 in. mercury and 60 deg. fahr	1,958,691	498,628
Difference	2,691	1,128
Per cent of difference	0.137	0.224

TESTS AGAINST WET METERS[1]

53 A Thomas meter was imported into Germany to be used in some scientific investigations of blowers and compressors. It was tested against a carefully calibrated wet meter at the Berlin IV Precinct Gas Works under the direct supervision of their engineers, who are of recognized technical ability and who exercised the most painstaking care throughout the tests. A few representative results of the Berlin tests are given below:

Duration of test, hours	1½	3½	3½
Cubic meters of gas by wet meter at 15.5 deg. cent. and 760	5410	7111	14600
Cubic meters of gas by Thomas meter	5400	7076	14537
Difference	10	35	63
Per cent difference	0.20	0.49	0.46

TESTS AT THE WORKS OF THE CUTLER-HAMMER MANUFACTURING COMPANY

54 Experimental work is constantly being carried on by the manufacturers of the Thomas meter. Two of the most interesting tests conducted there are described here through the courtesy of the Cutler-Hammer Manufacturing Company, as they demonstrate the accuracy of the Thomas meter under varying conditions.

55 *First Test.* Two automatically operated Thomas meters, one of 25,000 cu. ft. per hour capacity and the other of 50,000 cu. ft. per hour capacity, were put in the same pipe line in series. When air was flowing through the pipe it was noted that each meter recorded the same amount of air. By means of an electric heater in the pipe between the two meters the temperature of the air entering the second meter was gradually increased until it was 60 deg. fahr. higher than the temperature of the air entering the first meter. The amounts recorded on each meter were meanwhile carefully watched and it was observed that the readings still remained practically identical.

56 *Second Test.* A manually operated test meter was connected in a *horizontal* position in series with an automatically controlled meter which was in a vertical position, with a right angled bend intervening between them. This arrangement was made purposely to prevent the air passing through the two meters in the same manner. The test meter was manually operated on both 110 and 220 volts direct current and the automatic meter on 220 volts alternating current. The automatic meter had a capacity of 500,000 cu. ft. of free gas per hour. The results at different rates of flow were as follows:

9 per cent of maximum flow, error in automatic meter	+0.2 per cent
42 per cent of maximum flow, error in automatic meter	+0.2 per cent
81 per cent of maximum flow, error in automatic meter	+0.0 per cent

[1] Carl C. Thomas. Proc. Am. Gas Inst., vol. 7, 1912.

APPENDIX NO. 2

DATA AND CALCULATIONS

57 In order to show the manner in which the results of the calculations upon the pitot tubes were tabulated, Tables 16 to 18, applying to tubes *A*, *H* and *X* are here reproduced. The explanation of the calculations is as follows:

58 In column 1, Test Number, the first letter designates the pitot tube; the middle figure, the test in the series arranged according to the fan speed; and the last symbol, *F* or ½, signifies full gate or half gate at the discharge end of the test pipe.

59 Column 2 gives the average revolutions per minute of the fan as obtained by a hand revolution counter.

60 Column 3 shows the barometer reading in inches of mercury.

61 In column 4, the atmospheric pressure in pounds per sq. in. is obtained by multiplying the values in column 3 by the weight of a cu. in. of mercury taken from Chart *F* corresponding to room temperature.

62 Column 5 indicates the pressure in the pipe above atmospheric pressure at the point where the pitot tube was inserted.

63 In column 6, the pressure inside the pipe above the atmospheric pressure is the product of the values given in column 5, the specific gravity of the gasolene and the weight of the water per cu. in. taken from Chart *G* corresponding to room temperature.

64 In column 7, the absolute pressure in lb. per sq. in. on the air flowing through pipe at the pitot tube is the sum of the pressures given in column 4 and column 6.

65 Columns 8, 9 and 10 give the averages of the observed temperatures in deg. fahr.

66 In column 11, the percentage of humidity of the flowing air is obtained from Chart *A* and the temperatures given in columns 9 and 10.

67 In column 12, the weight of air partially saturated with water vapor in lb. per cu. ft. is obtained from Charts *D* and *E*. Knowing the temperature of the air (column 10) and the percentage of humidity (column 11) the weight of a cubic foot of air at 14 lb. per sq. in. pressure may be read from Chart *D*. To this value add the correction obtained from Chart *E* corresponding to the pressures given in column 7 and the temperature in column 10. The sum is the weight of a cubic foot of the mixture.

68 The calculations involved in finding the weight of a cubic foot of air partially saturated with water vapor are outlined in Appendix No. 3 and are based on well-known thermodynamic relations.

69 In column 13, the specific heat of the mixture of air and water vapor in B.t.u. per lb. is obtained from Chart *B* corresponding to the tem-

TABLE 16 TABULATED CALCULATIONS, PITOT TUBE A

Test No.	R.p.m. of Fan	Pressures: Barometer: In. of Mercury	Pressures: Barometer: Lb. per Sq. In.	Pressures: Static: In. of Gasolene, Sp. Gr.	Pressures: Static: Lb. per Sq. In.	Pressures: Absolute Lb. per Sq. In.	Temperatures: Room, Deg. Fahr.	Temperatures: Wet Bulb, Deg. Fahr.	Temperatures: Dry Bulb, Deg. Fahr.	Humidity, per cent	Weight of Air, Lb. per Cu. Ft.	Specific Heat of Air B.t.u. per Lb.	Thomas Meter: Calibration Constant, K	Thomas Meter: Amperes Corrected	Thomas Meter: Volts Corrected	Thomas Meter: Cu. Ft. of Air per Min., C_1	Pitot Tube: Velocity Heads: In. Gasolene, Sp. Gr. 0.733: Average Using Pitot Static, H_1	Pitot Tube: Velocity Heads: In. Gasolene, Sp. Gr. 0.733: Average Using Piezometer Static, H_2	Pitot Tube: Velocity Heads: In. Gasolene, Sp. Gr. 0.733: Center Using Pitot Static, H_3	Pitot Tube: Velocity Heads: In. Gasolene, Sp. Gr. 0.733: Center Using Piezometer Static, H_4	Pitot Tube: Velocity Heads: Ft. of Air: Average Using Pitot Static, h_1	Pitot Tube: Velocity Heads: Ft. of Air: Average Using Piezometer Static, h_2	Pitot Tube: Cu. Ft. of Air per Min.: Using Pitot Static C_2	Pitot Tube: Cu. Ft. of Air per Min.: Using Piezometer Static, C_3
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
A-1-½	794	29.200	14.267	1.17	0.031	14.298	80.0	69.8	84.0	48.5	0.07049	0.2445	0.02835	12.49	52.7	1080	0.1336	0.1434	0.1725	0.1765	7.22	7.75	1030	1068
A-2-½	875	29.206	14.270	1.37	0.036	14.307	81.5	70.9	85.1	49.0	0.07036	0.2446	0.02839	12.98	55.15	1178	0.1629	0.1723	0.2110	0.2170	8.82	9.34	1138	1171
A-3-½	951	29.210	14.272	1.41	0.037	14.309	82.2	71.4	86.3	48.5	0.07021	0.2446	0.02843	13.73	57.8	1315	0.1928	0.2060	0.2500	0.2650	10.47	11.18	1240	1280
A-4-½	1028	29.210	14.272	1.69	0.045	14.317	83.1	71.8	87.1	48.0	0.07013	0.2447	0.02848	14.33	60.05	1420	0.2355	0.2450	0.2970	0.3095	12.80	13.32	1371	1399
A-5-½	1107	29.212	14.273	1.96	0.052	14.325	84.0	72.9	88.4	48.0	0.06999	0.2448	0.02853	14.88	62.75	1552	0.2645	0.2860	0.3520	0.3670	14.40	15.57	1452	1510
A-6-½	1190	29.212	14.273	2.28	0.060	14.333	85.7	74.2	90.7	46.5	0.06969	0.2450	0.02865	15.43	64.75	1673	0.3087	0.3440	0.3980	0.4245	16.88	18.81	1573	1662
A-7-½	1266	29.210	14.272	2.60	0.069	14.341	86.8	74.8	91.7	46.0	0.06961	0.2451	0.02871	15.99	67.05	1803	0.3447	0.3893	0.4510	0.4880	18.87	21.31	1663	1770
A-8-½	1371	29.210	14.272	3.00	0.079	14.351	87.8	75.0	93.1	43.5	0.06949	0.2452	0.02878	16.49	69.75	1942	0.3956	0.4495	0.5150	0.5585	21.68	24.65	1782	1902
A-9-½	1447	29.208	14.271	3.38	0.089	14.361	88.5	75.2	94.0	41.5	0.06945	0.2452	0.02883	16.99	71.20	2049	0.4450	0.5122	0.5860	0.6360	24.40	28.10	1892	2030
A-1-F	795	29.142	14.239	0.18	0.005	14.244	86.0	77.2	89.2	58.0	0.06933	0.2456	0.02860	15.52	65.7	1710	0.3392	0.3508	0.4380	0.4440	18.64	19.28	1655	1680
A-2-F	870	29.150	14.243	0.22	0.006	14.249	86.5	76.9	89.8	56.0	0.06928	0.2455	0.02862	16.37	69.4	1910	0.4135	0.4335	0.5405	0.5505	22.74	23.85	1828	1870
A-3-F	947	29.148	14.242	0.27	0.007	14.249	87.0	77.1	90.5	54.5	0.06923	0.2455	0.02865	17.07	72.6	2085	0.4935	0.5235	0.6430	0.6635	27.15	28.80	1995	2058
A-4-F	1031	29.150	14.243	0.32	0.008	14.251	88.0	77.2	91.2	53.5	0.06912	0.2456	0.02870	17.72	75.7	2265	0.5780	0.6180	0.7520	0.7880	31.85	34.06	2160	2237
A-5-F	1118	29.148	14.242	0.37	0.010	14.252	88.8	78.0	92.8	52.0	0.06890	0.2457	0.02877	18.67	79.5	2487	0.6770	0.7365	0.8810	0.9310	37.43	40.73	2340	2440
A-6-F	1191	29.148	14.242	0.41	0.011	14.253	89.7	78.5	93.4	51.5	0.06882	0.2458	0.02879	19.22	81.9	2675	0.7780	0.8480	1.0040	1.0695	43.05	46.93	2513	2620
A-7-F	1273	29.150	14.243	0.46	0.012	14.255	90.0	78.9	94.3	51.0	0.06872	0.2459	0.02885	19.97	84.6	2885	0.8865	0.9685	1.1465	1.2290	49.16	53.65	2685	2805
A-8-F	1370	29.148	14.243	0.54	0.014	14.256	90.0	79.0	94.8	50.0	0.06867	0.2459	0.02887	20.67	87.4	3095	1.0220	1.1110	1.3100	1.4135	56.70	61.65	2880	3005
A-9-F	1458	29.130	14.233	0.62	0.016	14.249	91.0	79.6	96.3	48.5	0.06847	0.2458	0.02893	21.27	90.2	3300	1.1640	1.2815	1.4790	1.6070	64.75	71.30	3080	3240

peratures given in column 10 and the percentage of humidity given in column 11. The method of calculating specific heat of a mixture of air and water vapor for any temperature, pressure and per cent humidity is shown in Appendix No. 3.

70 In column 14, the value of K depends upon the calibration of the resistance thermometers in the Thomas meter and is taken from the curve

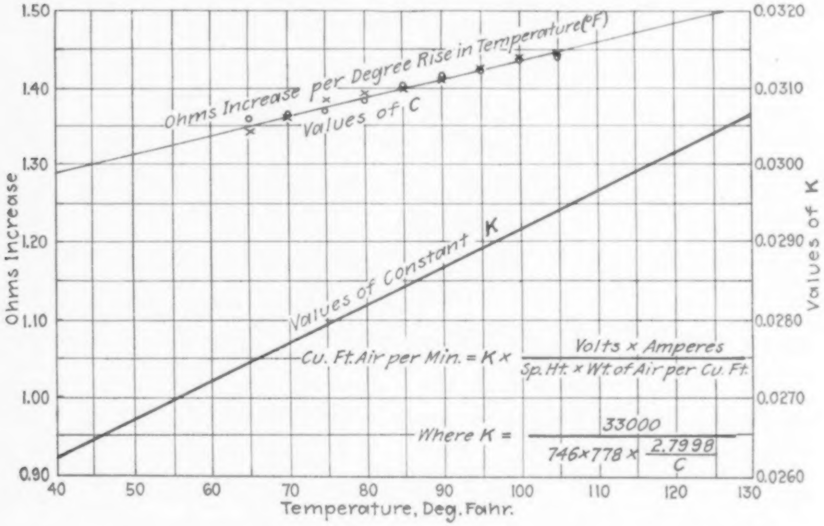


FIG. 26 CALIBRATION OF RESISTANCE THERMOMETERS IN THOMAS ELECTRIC METER

given in Fig. 26. It is defined by its use in the following formula, referred to later under column 17:

$$\text{Cu. ft. of air per minute by the Thomas meter} = K \frac{(\text{volts})\ (\text{amperes})}{(\text{specific heat})\ (\text{wt. air per cu. ft.})}$$

where

$$K = \frac{33000}{746 \times 778 \times \dfrac{2.7998}{C}}$$

See also the description of the Thomas meter in Pars. 13 to 17.

71 Columns 15 and 16 show the corrected readings of the voltmeter and ammeter which measured the electrical energy consumed by the heater in the Thomas meter.

72 In column 17, the cubic feet of air per minute as measured by the Thomas meter is calculated from the following formula:

$$C_1 = K \frac{(V)\ (A)}{S \quad W}$$

where

C_1 = cu. ft. of air per min. as measured by the Thomas meter

TABLE 17 TABULATED CALCULATIONS, PITOT TUBE X

Test No.	R.p.m. of Fan	Pressures: Barometer: In. of Mercury	Pressures: Barometer: Lb. per Sq. In.	Pressures: Static: In. of Gasolene, Sp. Gr.	Pressures: Static: Lb. per Sq. In.	Pressures: Absolute Lb. per Sq. In.	Temperatures: Room, Deg. Fahr.	Temperatures: Wet Bulb, Deg. Fahr.	Temperatures: Dry Bulb, Deg. Fahr.	Humidity, per Cent	Weight of Air, Lb. per Cu. Ft.	Specific Heat of Air B.t.u. per Lb.	Thomas Meter: Calibration Constant, K	Thomas Meter: Amperes Corrected	Thomas Meter: Volts Corrected	Thomas Meter: Cu. Ft. of Air per Min., C_1	Pitot Tube: Velocity Heads: In. Gasolene, Sp.Gr. 0.740–43: Average Using Pitot Static, H_1	Average Using Piezometer Static, H_2	Center Using Pitot Static, H_3	Center Using Piezometer Static, H_4	Ft. of Air: Average Using Pitot Static, h_1	Average Using Piezometer Static, h_2	Cu. Ft. of Air per Min.: Using Pitot Static C_2	Using Piezometer Static, C_3
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
X-1-½	785	29.100	14.200	0.90	0.024	14.224	95.0	83.0	98.5	52.5	0.06792	0.2465	0.02905	11.94	50.51	1047	0.1102	0.1293	0.1495	0.1685	6.25	7.33	958	1037
X-2-½	876	29.078	14.185	1.12	0.030	14.215	95.8	81.8	99.0	48.0	0.06788	0.2463	0.02907	12.51	53.32	1161	0.1348	0.1596	0.1795	0.2025	7.66	9.07	1060	1153
X-3-½	954	29.058	14.177	1.35	0.036	14.213	96.0	81.5	99.0	47.5	0.06787	0.2462	0.02907	13.16	55.83	1270	0.1603	0.1901	0.2115	0.2410	9.10	10.80	1156	1260
X-4-½	1028	29.016	14.165	1.60	0.042	14.207	96.0	80.9	99.1	46.0	0.06785	0.2461	0.02908	13.68	57.92	1378	0.1896	0.2287	0.2510	0.2940	10.75	12.98	1256	1380
X-5-½	1120	29.028	14.170	1.86	0.049	14.219	95.5	80.9	99.7	44.0	0.06785	0.2461	0.02911	14.23	60.02	1490	0.2230	0.2659	0.2950	0.3460	12.66	15.08	1363	1488
X-6-½	1195	29.054	14.177	2.16	0.057	14.234	96.0	81.6	100.6	44.5	0.06780	0.2462	0.02915	14.73	62.05	1595	0.2548	0.3106	0.3360	0.3930	14.46	17.63	1457	1609
X-7-½	1288	29.066	14.180	2.55	0.068	14.248	96.3	81.8	100.9	44.5	0.06785	0.2463	0.02917	15.23	64.45	1713	0.2918	0.3568	0.3840	0.4570	16.55	20.25	1557	1723
X-8-½	1370	29.066	14.180	2.92	0.077	14.257	96.3	81.8	101.0	44.5	0.06786	0.2463	0.02918	15.79	66.58	1834	0.3354	0.4192	0.4380	0.5305	19.03	23.80	1670	1870
X-9-½	1460	29.070	14.181	3.30	0.088	14.269	96.5	82.0	101.5	44.0	0.06785	0.2463	0.02920	16.31	68.72	1960	0.3792	0.4788	0.4940	0.6035	21.50	27.20	1775	1996
X-1-F	807	29.034	14.173	0.18	0.005	14.178	80.2	73.3	83.6	62.2	0.06977	0.2452	0.02834	15.03	64.55	1605	0.2840	0.3395	0.3565	0.4295	15.55	18.58	1510	1650
X-2-F	888	29.026	14.168	0.22	0.006	14.174	80.5	74.0	84.1	62.2	0.06967	0.2453	0.02835	15.80	68.75	1800	0.3404	0.4142	0.4290	0.5220	18.70	22.72	1655	1825
X-3-F	963	29.015	14.160	0.26	0.007	14.167	81.0	75.0	85.0	63.0	0.06950	0.2453	0.02838	16.64	70.80	1962	0.4049	0.4968	0.5100	0.6240	22.25	27.30	1805	2000
X-4-F	1046	29.014	14.160	0.31	0.008	14.168	81.5	75.4	85.7	63.0	0.06940	0.2454	0.02842	17.35	73.82	2140	0.4875	0.5874	0.6155	0.7455	26.65	32.35	1975	2180
X-5-F	1135	29.017	14.161	0.38	0.010	14.171	82.0	76.0	86.2	63.0	0.06935	0.2455	0.02845	18.15	77.38	2345	0.5585	0.7014	0.7270	0.8840	30.75	38.60	2125	2380
X-6-F	1229	29.015	14.160	0.40	0.011	14.171	82.5	76.2	86.9	62.7	0.06926	0.2456	0.02848	18.96	80.20	2545	0.6433	0.8124	0.8320	1.0250	35.50	44.80	2280	2562
X-7-F	1311	29.007	14.156	0.47	0.013	14.169	83.3	76.7	88.0	60.5	0.06912	0.2457	0.02853	19.65	83.27	2745	0.7451	0.9279	0.9650	1.1920	41.20	51.25	2455	2740
X-8-F	1410	29.006	14.155	0.52	0.014	14.169	84.0	77.3	89.0	59.5	0.06897	0.2457	0.02857	20.32	85.90	2940	0.8496	1.0863	1.0900	1.3590	47.05	60.20	2625	2970
X-9-F	1496	29.004	14.154	0.60	0.016	14.170	84.8	78.0	90.1	58.5	0.06885	0.2458	0.02862	21.01	88.70	3145	0.9560	1.2331	1.2290	1.5430	53.00	68.45	2785	3165

$(V)(A)$ = volts (column 15) times amperes (column 16) = watts consumed by electrical heater in the Thomas meter

S = specific heat of the air flowing (column 13)

W = weight of the air flowing in lb. per cu. ft. (column 12)

K is taken from column 14

73 Column 18 gives the mean velocity head as measured by the pitot tube using the pitot tube static pressure. It was obtained as follows: Readings were taken of the velocity head at 20 points on the cross-section of the pipe as described in Par. 21 and by Fig. 6. The square roots of these 20 readings were averaged and the square of this average is the value entered in column 18.

74 Column 19 indicates the mean velocity head as measured by the pitot dynamic tube and the piezometer static pressure. It was obtained in the manner described above for column 18.

75 Columns 20 and 21 record the velocity heads when the pitot tube is at the center of the pipe, column 20 using the pitot tube static pressure and column 21 using the piezometer static pressure.

76 Columns 22 and 23 are the velocity heads given in columns 18 and 19 reduced to feet of air flowing. They were calculated as follows:

$$h = 144\ spH$$

where

h = velocity head in ft. of air

s = specific gravity of gasolene

p = weight of water in pounds per cu. in., taken from Chart G

H = velocity head in in. of gasolene as given in columns 18 or 19

77 Column 24 gives the cubic feet of air per minute as measured by the pitot tube using the pitot tube static pressure.

78 Column 25 gives the cubic feet of air per minute as measured by the pitot dynamic tube and the piezometer static pressure.

79 The method of calculating the results given in columns 24 and 25 is as follows:

$$\text{for pitot tube } C_2 = 60\,A\,\sqrt{2gh_1} = 383\,\sqrt{h_1}$$

$$\text{for Stauscheibe } C_2 = 60\,A\,\frac{\sqrt{2gh_1}}{1.17} = 327\,\sqrt{2gh_1}$$

$$\text{for both } C_3 = 60\,A\,\sqrt{2gh_2} = 383\,\sqrt{h_2}$$

where

C_2 = cu. ft. of air per minute as measured by pitot tube using pitot tube static pressure

C_3 = cu. ft. of air per minute as measured by pitot dynamic tube and piezometer static pressure

A = area of cross-section of pipe where pitot tube was inserted = 0.7959 sq. ft.

g = 32.2

h_1 = velocity head in ft. of air taken from column 22 and described previously

h_2 = velocity head in ft. of air taken from column 23 and described previously

TABLE 18 TABULATED CALCULATIONS, PITOT TUBE *H*

Test No.	R.p.m. of Fan	Pressures: Barometer: In. of Mercury	Pressures: Barometer: Lb. per Sq. In.	Pressures: Static: In. of Gasolene, Sp. Gr.	Pressures: Static: Lb. per Sq. In.	Pressures: Absolute Lb. per Sq. In.	Temperatures: Room, Deg. Fahr.	Temperatures: Wet Bulb, Deg. Fahr.	Temperatures: Dry Bulb, Deg. Fahr.	Humidity, per Cent	Weight of Air, Lb. per Cu. Ft.	Specific Heat of Air B.t.u. per Lb.	Thomas Meter: Calibration Constant, K	Thomas Meter: Amperes Corrected	Thomas Meter: Volts Corrected	Thomas Meter: Cu. Ft. of Air per Min., C_1	Pitot Tube: Velocity Heads: In. Gasolene, Sp. Gr. 0.745: Average Using Pitot Static, H_1	Pitot Tube: Velocity Heads: In. Gasolene, Sp. Gr. 0.745: Average Using Piezometer Static, H_2	Pitot Tube: Velocity Heads: In. Gasolene, Sp. Gr. 0.745: Center Using Pitot Static, H_3	Pitot Tube: Velocity Heads: In. Gasolene, Sp. Gr. 0.745: Center Using Piezometer Static, H_4	Pitot Tube: Velocity Heads: Ft. of Air: Average Using Pitot Static, h_1	Pitot Tube: Velocity Heads: Ft. of Air: Average Using Piezometer Static, h_2	Pitot Tube: Cu. Ft. of Air per Min.: Using Pitot Static C_2	Pitot Tube: Cu. Ft. of Air per Min.: Using Piezometer Static, C_3
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
H-1-½	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
H-2-½	870	29.182	14.270	1.22	0.032	14.302	73.5	63.0	79.0	40.5	0.07129	0.2437	0.02808	13.33	55.93	1203	0.1718	0.1737	0.2180	0.2190	9.33	9.43	1170	1176
H-3-½	948	29.182	14.270	1.46	0.039	14.309	73.5	63.8	80.2	40.0	0.07127	0.2438	0.02814	13.91	58.28	1313	0.2039	0.2067	0.2630	0.2650	11.05	11.22	1274	1284
H-4-½	1029	29.182	14.270	1.70	0.045	14.315	76.0	65.3	82.1	40.0	0.07092	0.2439	0.02824	14.46	60.42	1425	0.2410	0.2459	0.3065	0.3125	13.14	13.40	1390	1403
H-5-½	1110	29.182	14.270	2.05	0.054	14.324	77.5	66.0	83.3	39.5	0.07080	0.2439	0.02830	15.20	63.15	1570	0.2900	0.2943	0.3610	0.3720	15.80	16.05	1523	1535
H-6-½	1185	29.182	14.270	2.34	0.062	14.332	78.9	66.8	84.6	39.0	0.07066	0.2440	0.02836	15.66	65.52	1685	0.3284	0.3407	0.4155	0.4280	17.97	18.62	1623	1654
H-7-½	1265	29.184	14.270	2.72	0.072	14.342	79.9	67.8	85.9	39.0	0.07052	0.2441	0.02844	16.15	67.42	1793	0.3774	0.3879	0.4745	0.4900	20.62	21.20	1738	1763
H-8-½	1360	29.186	14.270	3.08	0.082	14.352	81.5	69.5	88.0	39.0	0.07027	0.2443	0.02853	16.71	69.87	1932	0.4330	0.4516	0.5440	0.5640	23.80	24.79	1870	1910
H-9-½	1454	29.186	14.270	3.51	0.093	14.363	82.5	70.2	89.4	38.5	0.07013	0.2444	0.02860	17.24	72.15	2072	0.4881	0.5153	0.6200	0.6480	26.90	28.43	1986	2040
H-1-F	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
H-2-F	872	29.150	14.245	0.230	0.006	14.251	80.0	68.7	83.9	46.5	0.07029	0.2448	0.02833	16.74	70.24	1930	0.4579	0.4541	0.5700	0.5690	25.20	24.95	1922	1912
H-3-F	947	29.162	14.248	0.280	0.007	14.255	81.0	68.7	84.7	44.3	0.07021	0.2448	0.02837	17.47	73.28	2108	0.5592	0.5556	0.6735	0.6725	30.75	30.75	2123	2123
H-4-F	1024	29.168	14.252	0.310	0.008	14.260	82.1	69.2	86.3	42.0	0.07002	0.2449	0.02845	18.23	76.26	2300	0.6391	0.6363	0.8030	0.8055	35.27	35.10	2275	2269
H-5-F	1110	29.168	14.252	0.380	0.010	14.262	83.8	70.8	88.1	39.0	0.06981	0.2448	0.02854	18.96	79.47	2508	0.7500	0.7493	0.9435	0.9465	41.45	41.40	2465	2463
H-6-F	1195	29.168	14.252	0.420	0.011	14.263	84.5	71.3	89.4	41.0	0.06962	0.2449	0.02860	19.70	82.24	2710	0.8670	0.8651	1.0940	1.0950	48.10	48.00	2656	2653
H-7-F	1273	29.168	14.252	0.510	0.013	14.265	85.8	72.3	90.8	41.0	0.06944	0.2453	0.02868	20.37	84.91	2910	0.9958	0.9955	1.2465	1.2490	55.45	55.45	2855	2855
H-8-F	1375	29.170	14.252	0.550	0.015	14.267	86.5	73.2	92.1	40.5	0.06927	0.2454	0.02873	21.02	88.00	3120	1.1498	1.1481	1.4380	1.4370	64.00	63.95	3062	3060
H-9-F	1456	29.168	14.252	0.630	0.017	14.269	87.5	74.0	93.4	40.0	0.06910	0.2455	0.02880	21.71	90.48	3330	1.3028	1.3097	1.6330	1.6300	72.70	73.00	3265	3270

APPENDIX NO. 3

CHARTS SHOWING WEIGHTS PER CUBIC FOOT AND SPECIFIC HEATS OF MIXTURES OF AIR AND WATER VAPOR

As explained in the paper, the accurate comparison of results obtained by the Thomas meter and the pitot tube depends upon the use of correct values of the properties of air. It was therefore necessary to make a thorough study of this subject, the results of which are presented in this appendix. The formulae and values were gathered from various sources and represent the most modern information in regard to mixtures of air and water vapor.

As the calculations involved are long and tedious, the author has devised and constructed charts which are here presented in the hope that they may be of value to others. The charts were originally drawn to a much larger scale on a tracing about 2 ft. wide by 5 ft. long. Blue prints from this tracing may be obtained from the Society for 25 cents each.

OUTLINE OF CALCULATIONS TO ACCOMPANY CHARTS D AND E

P_t = total pressure of mixture $= p_a + xp_w$ in lb. per sq. in.
p_a = pressure of dry air in lb. per sq. in.
p_w = saturated vapor pressure in lb. per sq. in. (Marks and Davis steam tables used).
x = per cent humidity.
t = temperature of air in deg. fahr.
W = weight of cu. ft. of a mixture of air and water vapor at t temperature; P_t pressure; and x per cent humidity
w_a = weight of cu. ft. of dry air at a pressure of $(14.0 - xp_w)$ lb. per sq. in.
w_c = correction to be added to w_a for pressures above 14.0 lb. per sq. in.
w_w = weight of water vapor contained in 1 cu. ft. of saturated air
S = specific heat of a mixture of air and water vapor (B.t.u. per lb.)
S_a = specific heat of dry air $= 0.24112 + 0.000009\ t$ (Harvey N. Davis, Trans. Am. Soc. M. E., vol. 30, p. 750, 1908.)
S_w = specific heat of water vapor $= 0.4423 + 0.00018\ t$ (Willis H. Carrier, Trans. Am. Soc. M. E., vol. 33, p. 1016, 1911.)
R = 53.35
T = $459.6 + t$

For Dry Air:

$$p_a V_a = RT$$

$$w_a = \frac{1}{V_a} = \frac{p_a}{RT}$$

$$W = w_a + xw_w + w_c$$

$$w_a = \frac{(144)\ (14.0 - xp_w)}{(53.35)\ (459.6 + t)}$$

$$w_c = \frac{(144)\ [P_t - 14.0]}{(53.35)\ (459.6 + t)}$$

$$S = \frac{(w_a + w_c)\ (S_a) + (xw_w)\ (S_w)}{w_a + xw_w + w_c}$$

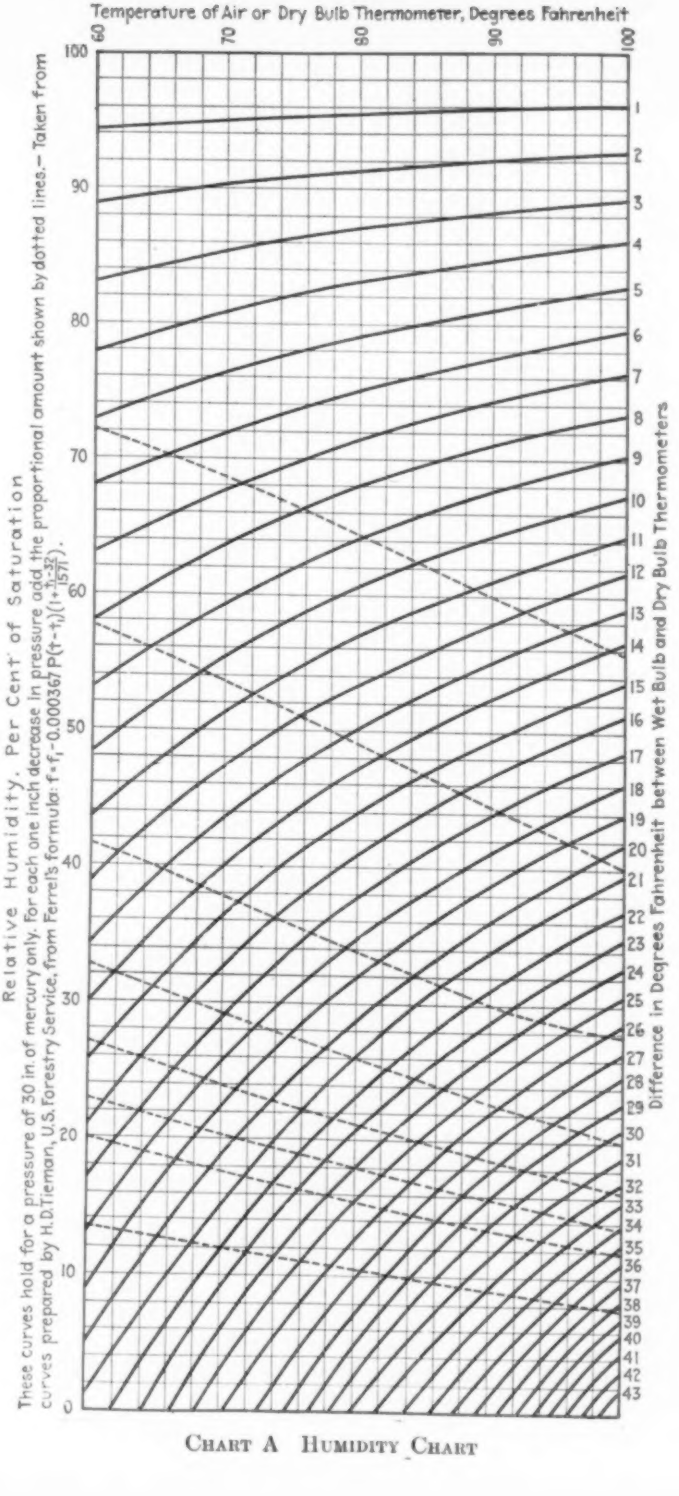


CHART A HUMIDITY CHART

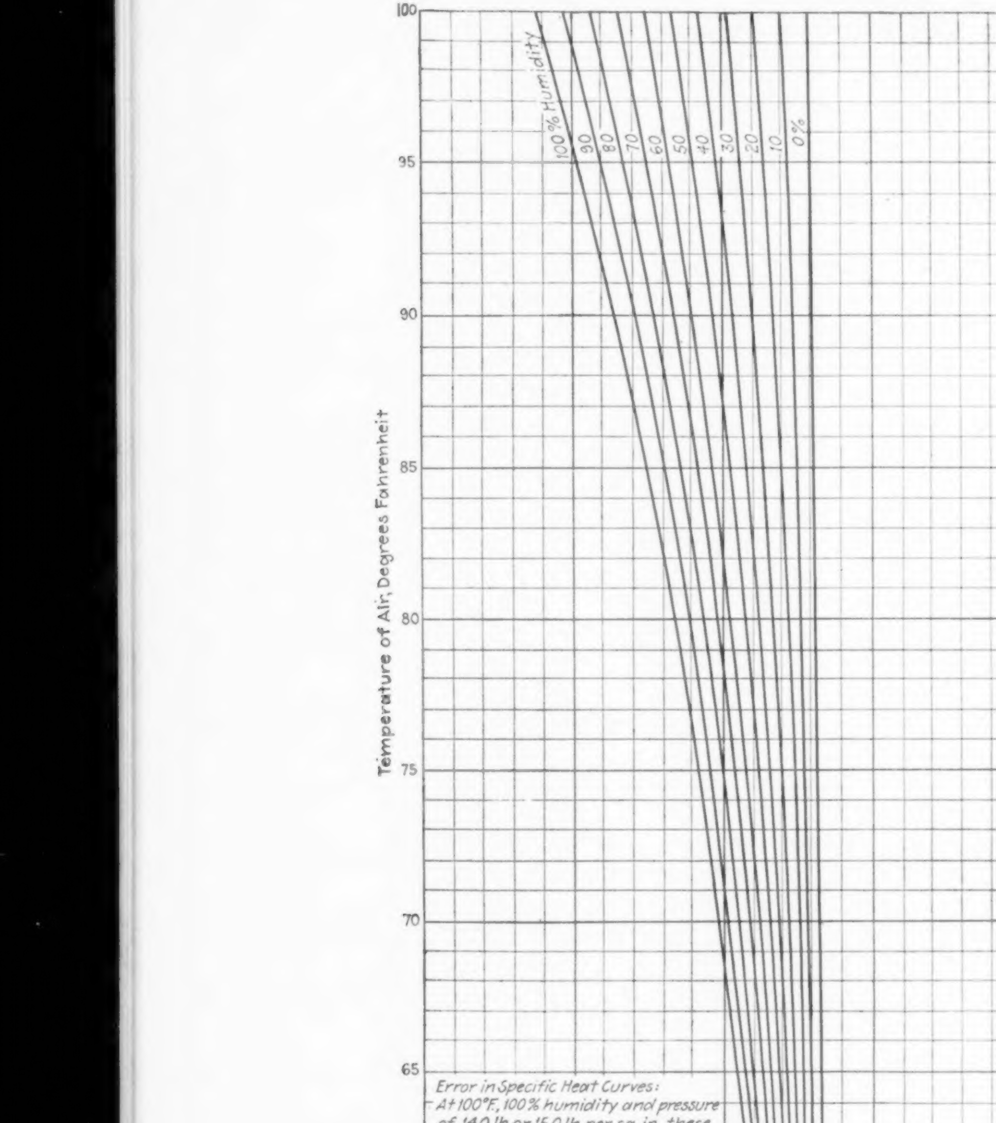


Chart B Chart giving the Specific Heat of Mixtures of Air and Water Vapor at 14.5 Lb. per Sq. In. Absolute Pressure for Varying Conditions of Temperature and Humidity

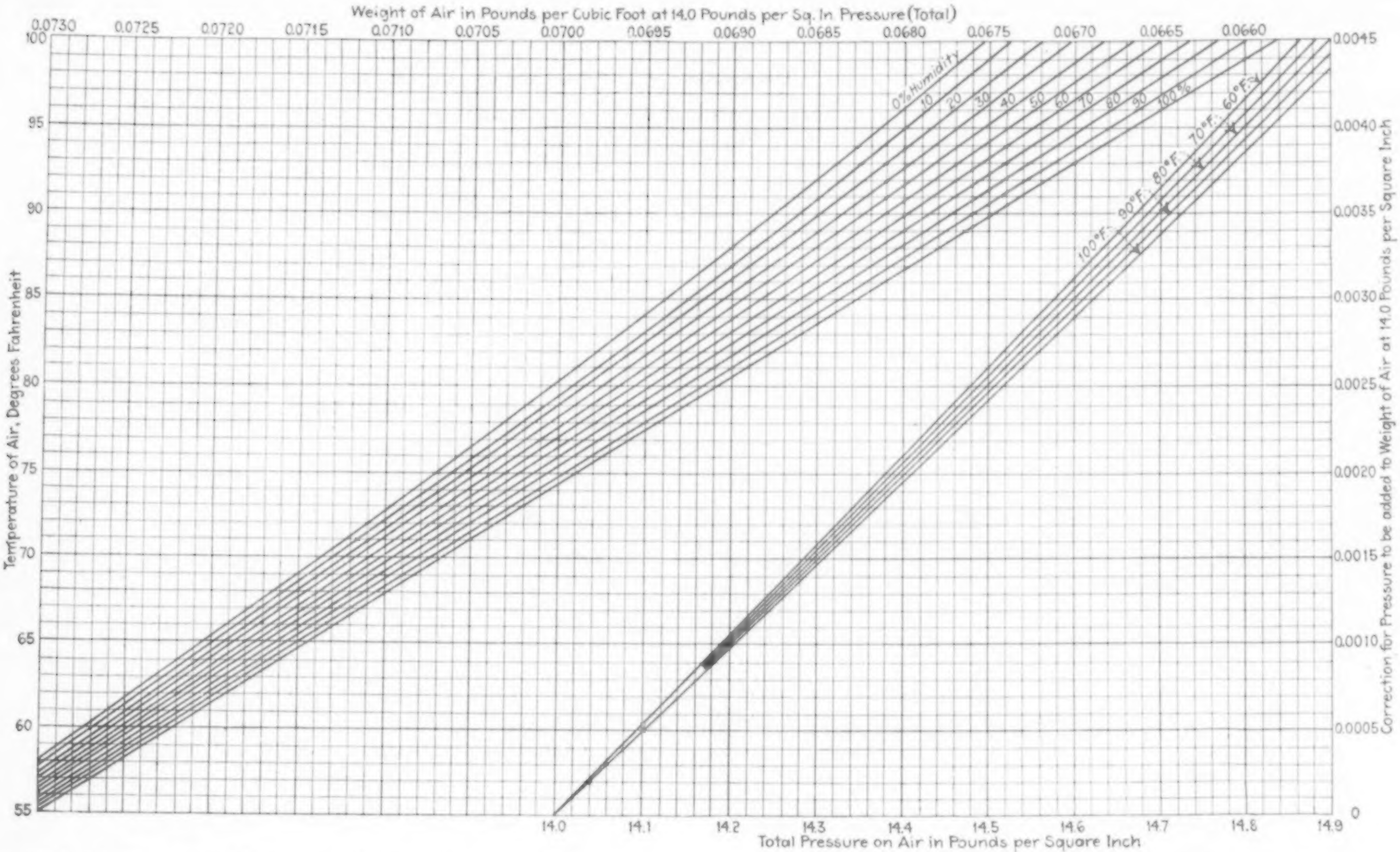


CHART D Chart giving the Weight of Mixtures of Air and Water Vapor at 14.0 Lb. per Sq. In. Absolute Pressure for Varying Conditions of Temperature and Humidity

CHART E Chart giving the Values to be added to the Values Taken from Chart D for Absolute Pressures above 14.0 Lb. per Sq. In.

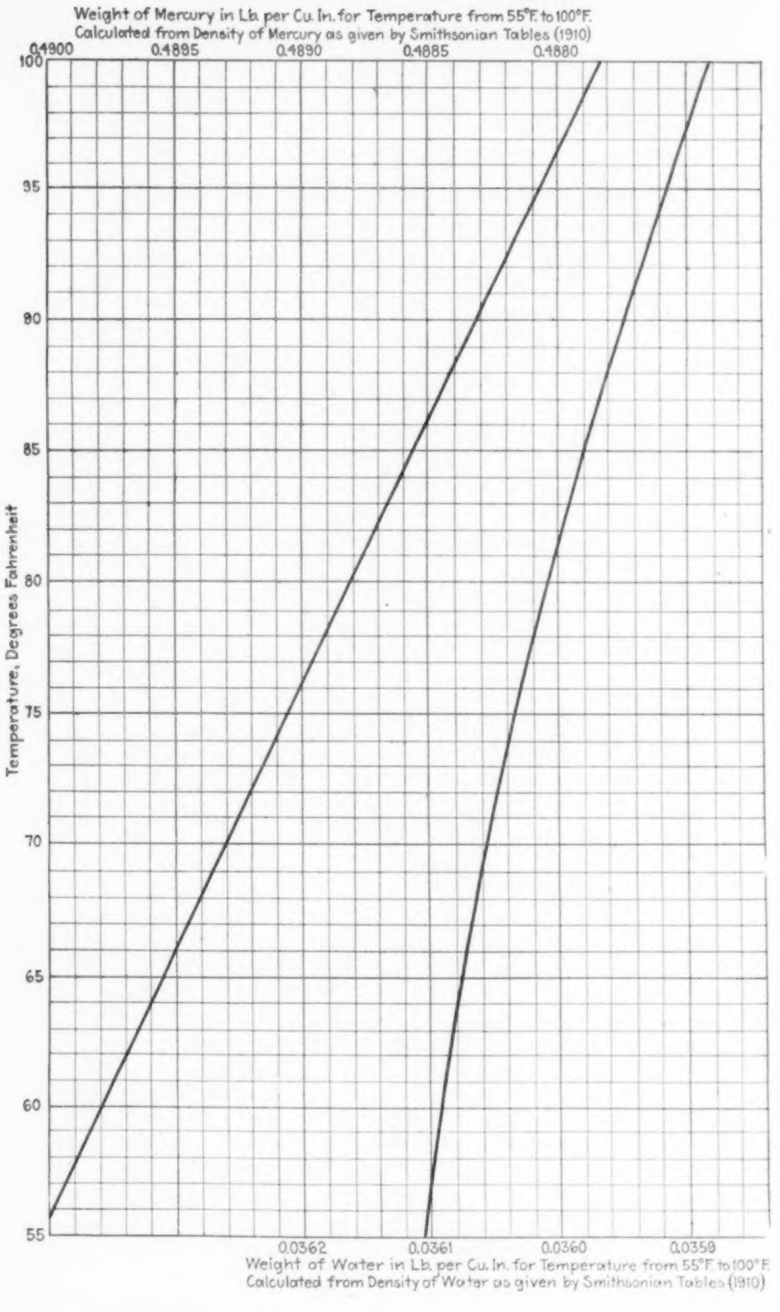


CHART F WEIGHT OF MERCURY IN POUNDS PER CUBIC INCH AT VARYING TEMPERATURES

CHART G WEIGHT OF WATER IN POUNDS PER CUBIC INCH AT VARYING TEMPERATURES

TESTS OF VACUUM CLEANING SYSTEMS

By J. R. McColl

ABSTRACT OF PAPER

The paper gives an outline of some tests of vacuum cleaners made for the Board of Education, Detroit, Mich. The tests were divided into two series: (*a*) with the machines on hose and piping as presented or recommended by the manufacturers; and (*b*) with all machines on the same hose and piping system, for the purpose of direct comparison. Each of these series was divided into two parts: (1) tests for ability to do work at the tool end of the hose; (2) analyses of machines as machines.

The paper outlines some of the fundamental principles of vacuum cleaning and contains various curves and tabulated data giving the results of the tests in detail, covering energy developed, power consumption, losses in hose and piping, efficiency, etc.

TESTS OF VACUUM CLEANING SYSTEMS

By J. R. McColl, Detroit, Mich.

Member of the Society

In June 1911, the firm of engineers of which the writer is a member conducted a series of tests on vacuum cleaning systems for the Board of Education of Detroit, Mich., for the purpose of comparing the merits of the various stationary machines for which proposals had been received, to be used in equipping eight school buildings. The specifications provided that the vacuum cleaning system must be of a capacity such as to take care of two sweepers operating simultaneously on a given system of piping, each sweeper to be provided with 75 ft. of hose. The capacity requirement for each sweeper was stipulated as 80 cu. ft. of free air to be handled with a vacuum of 1 in. of mercury inside an orifice at the tool or sweeper end of the hose. These requirements or specifications for volume, vacuum, and equivalent working orifice, at the end of the prescribed hose, any two of which settle the third, represent what was specified as "the ability to do work."

2 The specifications provided that all bidders must submit their machines for two series of tests: *A*, with the piping system and hose as proposed or required by the manufacturer, for the purpose of determining if under these conditions, the machine would give the required volume and vacuum as above outlined; *B*, with an arbitrary hose and piping system which would be the same for all machines, in order to make a direct comparison of the various machines under identical service conditions. The importance of this second series was emphasized by the specifications. Each of the two series of tests was divided into two parts: (1) measuring the "ability to do work" at the tool end of the prescribed hose; (2) analysis of the machine from the standpoints

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

of maintenance and repairs, simplicity, efficiency of lubrication, dust separation and collection, floor space, noise, tools and hose. Separate grades were given to each machine on its ability to do work and on its character as a machine, and these grades were then combined to get the total score.

3 The specifications provided that in grading machines the ability to do work would not count for more than two-thirds of the total score. It was the sense of the committee who assisted that a machine which makes a fine showing in ability to do work and yet has a low score as a machine is as much to be avoided as one which gains a high score as a machine and shows a poor ability to do work. The two features must be considered together in rating a machine. After going over various details involved, the consensus of opinion was that for a school building, 60 points should be given to the machine as a machine and 40 points to its ability to do work. This was of course an arbitrary standard, regarding which there would no doubt be a wide diversity of opinion among engineers, but as long as all machines were scored uniformly in accordance with this assumed standard, it has the stamp of fairness. The 60 points for the machine as a machine, were divided as follows:

Maintenance and repairs	20	
Simplicity	5	
Lubrication	5	
Dust collection	5	
Space	5	
Noise	5	
Tools and hose	15	60
Ability to do work		40
		100

4 Regarding the possible question that may arise in the tests of Series *A* where the machines were not operating under similar conditions as to hose and piping, whether any scores should have been given, on the grounds that it is unfair to grade the results of a machine using a large hose with the results of one working on a small hose, it should be explained that the distinction between the two series of tests is as follows: in the first series, the hose and piping system as proposed was considered an integral part of the machine, inseparable from it; in Series *B* the hose and piping were treated as auxiliary to the machine and independent of it. The scores of Series *A* represent simply

the results obtained from the various machines as presented, though working under dissimilar conditions. The scores of Series *B* represent the comparative merits of the machines when put on the same hose and piping and working under the same conditions.

5 The total grades for the various machines made up from the results of the tests as to the ability to do work, combined with the rating of the machines as machines, are given in column 6, Table 1. The method of computation for that part of the grade covering the ability to do work was made up on the basis of 40 points for meeting full specification requirements of 80 cu. ft. of air per sweeper, with 1-in. vacuum inside the equivalent working orifice. Machines which exceeded this capacity were scored proportionally above 40; and those falling below this capacity were correspondingly debited. It is evident, therefore, that the total score is made up of the sum of the various scores covering the machine as a machine on the basis of 60 points, combined with its ability to do work on the basis of 40 points, when just fulfilling specified requirements. The scores are therefore not percentages, but measures of standing on a reasonable, though arbitrary, scale. The percentages basis could have been used, had no extra credit been given to machines exceeding the specified requirements, or if the capacity attainments of the highest machine were taken as the basis of 40 points and others graded accordingly, but the method used was simpler and equally fair.

6 It was outlined in the specifications that the piping system used as a basis to work from would be a 2-in. line for the inlet system and a 2½-in. for the exhaust system, but manufacturers were permitted to specify other sizes for the first series of tests as a proposed part of their equipment, if so desired or required. This piping system was furnished and installed by the Board of Education. One manufacturer, bidding on the fan type of machine, was permitted to install a 3-in. piping system for use with his machine. Series *B* of the tests included runs made for all high-vacuum machines on the 2-in. piping system, as well as for all machines on a 3-in. piping system, under identical conditions as to hose. It was the writer's expectation to make the comparative series of tests, outlined as Series *B*, with all machines on the largest piping system and the largest hose required or specified by any manufacturer, but he was unable to carry out the plan as regards the hose, and the 1¼-in. hose of the Board of Education was used for all comparative tests.

TABLE 1 RESULTS OF TESTS

1	2	3	4	5	6	7	8	9		10	11	12			
Designating Letter	Type of Machine	No. of Sweeper	Size of Piping. In.	Size of Hose, In.	Score	Electric Current Consumption in Kw-Hr.		Cu. Ft. Free Air for 1 In. Vacuum Inside Orifice		Vacuum at Machine for 80 Cu. Ft. Free Air per Minute per Hose	Apparent Displacement at Machine for 80 Cu. Ft. Free Air	Pull to Move Following Lengths of Hose on Floor in Lb.			
						For Column 9	For 80 cf. Fr. Air					75 Ft.	37½ Ft.	25 Ft.	12½ Ft.
		As proposed—one-sweeper capacity													
F_1	Rotary Exhauster......	1	2	1½	93.4	2.72	3.23	86.6	2nd test	7.13	106.0	18.25	8.25	5.5	3.0
D_1	Multi-Stage Fan.......	1	3	1½	93.2	1.85	1.76	90.5		3.75	92.0	22.5	9.5	7.0	3.5
G_1	Single-Stage Fan.......	1	3	25′ = 2¼″ + 50′ = 1¾	85.6	1.62	....	61.5		....	....	22.0	7.0	4.5	2.5
F_1	Rotary Exhauster......	1	2	1¼	84.0	2.15	....	67.3	1st test	....	....	13.0	5.5	4.75	2.5
C_1	Rotary Exhauster......	1	2	1¼	82.0	2.5	2.5	82.0		76.2	108.5	19.5	9.0	4.75	2.5
		As proposed—two-sweeper capacity													
G_2	Single-Stage Fan.......	2	3	50′ = 2½″ + 25′ = 1¾	97.6	1.95	.95	177.8		2.07	86.0	22.0	7.0	4.5	2.5
F_2	Rotary Exhauster......	2	2	1½	85.5	3.95	....	142.0	2nd test	....	....	18.25	8.25	5.5	3.0
B_2	Rotary Exhauster......	2	2	1¼	83.0	9.25	4.22	175.5	2nd test	9.87	121.0				
F_2	Rotary Exhauster......	2	2	1¼	78.7	3.99	....	115.3	1st test	....	....	13.0	5.5	4.75	2.5
B_2	Rotary Exhauster......	2	2	1	66.0	10.8	....	93.8	1st test	....	....	23.0	10.75	5.5	4.0
E_2	Reciprocating, Plunger.	2	2	1¼	57.7	11.5	5.37	169.0		9.55	118.8	19.5	9.0	4.75	2.5
A_2	Reciprocating, Plunger.	2	2	1	35.4	6.7	....	97.1		....	....	25.0	13.5	8.5	5.25

		Comparison on school board hose—one sweeper													
F_1	Rotary Exhauster......	1	3		1¼	90.3	3.11	3.1	80.8		7.3	107.0			
F_1	Rotary Exhauster......	1	2		1¼	88.9	3.27	3.0	78.2		....	....			
D_1	Multi-Stage Fan.......	1	3		1¼	86.9	1.65	....	59.0		....	....			
C_1	Rotary Exhauster......	1	3		1¼	80.5	2.45	2.5	78.8		7.25	106.5			
G_1	Single-Stage Fan.......	1	3		1¼	68.7	0.7	....	28.0		....	....			
		Comparison on school board hose—two sweepers													
F_2	Rotary Exhauster......	2	3		1¼	83.9	4.67	....	135.0		....	....			
B_2	Rotary Exhauster......	2	2		1¼	83.0	9.25	4.22	175.5		9.87	121.0			
F_2	Rotary Exhauster......	2	2		1¼	82.0	4.25	....	128.4		....	....			
B_2	Rotary Exhauster......	2	3		1¼	75.3	3.5	....	131.0		....	....			
G_2	Single-Stage Fan.......	2	3		1¼	72.3	1.45	....	69.5		....	....			
E_2	Reciprocating, Plunger.	2	3		1¼	61.5	12.73	6.5	183.0		10.1	122.5			
E_2	Reciprocating, Plunger.	2	2		1¼	57.7	11.45	5.37	168.6		9.55	118.8			
A_2	Reciprocating, Plunger.	2	3		1¼	48.5	5.4	....	148.7		....	....			
A_2	Reciprocating, Plunger.	2	2		1¼	45.8	5.3	....	139.0		....	....			
		3-in. Piping and proposed hose													
F_1	Rotary Exhauster......	1	3		1½	95.3	2.58	3.15	90.8		7.1	105.6			
F_2	Rotary Exhauster......	2	3		1½	86.3	3.82	....	146.0		....	....			
G_1	Single-Stage Fan.......	1	3	25′ = 2¼″ + 50′ =	1¾	85.6	0.78	....	61.5		....	....			
C_1	Rotary Exhauster......	1	3		1¼	80.5	2.45	2.5	78.8		7.25	106.5			
A_2	Reciprocating, Plunger.	2	3		1	34.4	6.9	....	94.0		....	....			

7 It is obvious that all machines would have given much better results, as regards ability to do work, and lower power consumption, had the comparative tests been made with a larger hose, but in the writer's opinion, as well as that of one or two of the committee who had much to do with air engineering, the general order of standings would not have been very materially changed, judging from the apparent displacements at the machine, as shown in column 12 of Table 1. The exact relative positions and grades, however, could be determined only by the actual tests.

8 Machines were submitted by seven manufacturers, two submitting both single-sweeper and two-sweeper outfits. A variety of machines was represented; there were two fan-type machines, one a single-stage and the other a multi-stage; two reciprocating plunger-type machines; and three of the rotary exhauster or impeller type. It was interesting to note that machines submitted for the same specified duty, ranged in weight from a few hundred pounds to a ton or more. Another interesting feature was that the vacuum at the machine in the effort to accomplish the same specific requirements as to volume at the tool end of the proposed hose, ranged from little more than 2 in. of mercury to approximately 15 in. The power consumption for the same ability to do work ranged from approximately $1\frac{3}{4}$ kw. to over 13 kw., depending upon the particular type of machine, and especially on the size of the piping and hose used, which ranged in size from 1 in. in diameter to $2\frac{1}{2}$ in. in diameter, the latter size being that at the inlet connection for a hose tapered down from $1\frac{3}{4}$ in. in diameter at the tool.

9 No uniformity in sizes of hose and tool handles was found among the manufacturers of so-called high-vacuum machines, and low-vacuum machines, and it will doubtless be a long time before they will agree on an exact size of hose and tool handle. These will always be largely matters of personal taste and also matters depending on the importance put by the manufacturer or user, on the question of convenience as contrasted with the importance of the greater ability to do work at a less consumption of power. The larger the hose and tool handle, within reasonable limits, the more rapid and effective work the operator should be able to do with the least consumption of power. But weight of hose and tool handles is often confounded with size. Some of the hose submitted with machines for the tests could have been

much lighter and yet have been strong enough for the maximum vacuum on it. The weight of the hose, for convenience sake, should be as light as possible, consistent with durability and the maximum vacuum desired or required to be carried on it, not only for its new rotund condition, but for its possible flattened condition after months of use.

10 As regards the size of piping, the larger this is, the higher the available vacuum for the hose; if too large, however, the average velocity of the air will not be sufficient to insure the carrying of refuse in horizontal runs, and to prevent clogging. Experience led to the belief that the velocity should not fall much below 2000 ft. per min. in a horizontal run. A tool under ordinary working conditions is handling a varying volume of air. Fortunately at frequent intervals the tool is lifted from the floor or tipped to a marked degree, thereby permitting a comparatively large volume of air to be handled for a brief period at least. If these periods are frequent enough, the volume of air thus intermittently handled will tend to keep the piping system clean, even though the average velocity may be below that already given. Vertical risers should not clog. The factors which operate to limit or prevent the use of large piping are: first cost of installation, the space occupied, the question of cutting into building walls, and the unsightly appearance of large pipes. On the other hand, the enormous wastes due to small hose and piping, so conspicuously brought out in some of the results of the tests, make it desirable to use as large piping as possible, to say nothing of the hose.

11 It seems reasonable to believe that the time will come when the maximum and minimum sizes of hose and piping systems will be standardized for various types of buildings and for various kinds of service, and that these features will be just as distinctly specified as are now the sizes of wires for a given electric service, or as the size of steam pipes and return lines for a given heating service.

METHODS OF TESTING

12 The Capron school in Detroit, one of the buildings to be equipped with a vacuum cleaning system, was used for the tests and all machines were delivered to the basement for this purpose. The 2-in. piping system in the building and the 3-in. piping system already mentioned were arranged so that each machine in

turn could be connected for the various tests, which covered power to drive, ability to do work, vacuum at the machine, speeds, etc.

13 For the work of the committee assisting the author in analyzing the machines as machines, and in grading each according to its merits on the basis of the arbitrary standards given above, each machine was taken into one of the basement rooms, dismantled and critically inspected from the various standpoints already

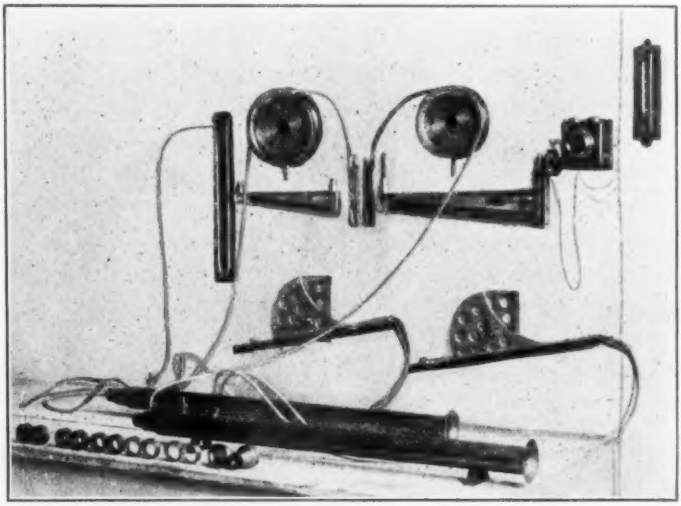


FIG. 1 VIEW OF APPARATUS FOR TESTING ABILITY TO DO WORK

outlined. Copious notes were made and the various points fully discussed and considered in the subsequent meeting where the final grades were made up.

14 For the power tests at the machine, the usual readings of speed, current consumption, vacuum, etc., were made, similar to those taken in other lines of tests.

15 The apparatus for measuring the ability to do work at the tool end of the hose was set up in the principal's office on the first floor and two lines of hose 75 ft. in length were carried from inlet connections on the first floor corridor to this room. As already outlined, the end of each hose was restricted by a series of arbitrary orifices, introduced successively, and the vacua with the corresponding volumes of air handled were determined.

16 The apparatus used in these determinations is illustrated in Fig. 1, which is a photograph taken immediately after the tests. The pitot tube method for determining air velocities was used in the measurement of air volumes. The type of pitot tube used was one which the writer and others who have had much to do in recent years with air measurements have found the most accurate. It was similar to the pitot tube described in the paper by Mr. Charles H. Treat.[1] Fig. 2 shows a cross-section of the rectifying barrel or tube, pitot and static tubes, orifice, and orifice holder with vacuum gage connection, etc. The rectifying tube consists of a cylinder with an inside diameter of 2 in. and a length of approximately 36 in. The inlet end is made with a bell-mouth to overcome in some degree the vena contracta effect of the entering air, and to assist in making sure that the air is moving practically in parallel lines when it encounters the pitot tube.

17 The pitot and static tubes were soldered together and made into an L-shaped instrument, pointing up-stream, and the connections for the manometer were separated for easy connection, as shown. The pitot tube leg was placed parallel to the axis of the rectifying barrel. The dimensions and details of this instrument are shown in Fig. 3. It was first intended to make the pitot tube adjustable to any position across the diameter of the rectifying barrel in order to take a series of readings at properly selected points, and average them as outlined in Mr. Treat's paper. This would give more accurate results as to average velocities, but when the multiplicity of readings for the great number of tests to be made and the limited time available were considered, this idea was abandoned. The average of a series of readings for 80 cu. ft. of air per minute was taken and then the pitot tube was set at the point where it gave the average reading when handling this volume of air. When handling other volumes, the pitot tube reading at this fixed position would not be strictly accurate for determining the average velocity.

18 The static tube consisted of a small brass tube soldered to the pitot tube, the up-stream end being sharpened to split the air and cause as little eddying round the instrument as possible. Both pitot and static tubes were made as small as practicable to give as little disturbance as possible in the air currents. But some eddying, of course, is present at best, and to make sure that

[1] Trans. Am.Soc.M.E., vol. 34, p. 1019.

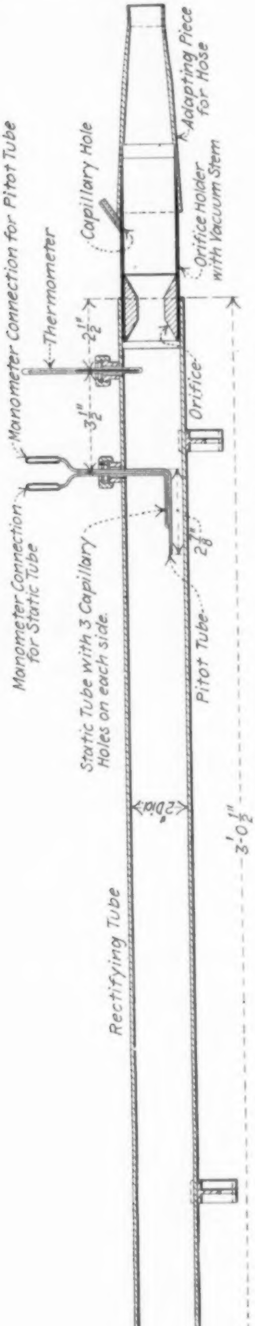


FIG. 2 SECTION OF PITOT TUBE APPARATUS AND ORIFICE HOLDER

this should not increase the true static reading, the perforations in the side walls of the static tube were made small capillary holes. Obviously the success of the pitot tube method of air measurement depends, in the first place, on the accuracy with which the pitot and static tubes will separate the pressure which is purely static, from that which is the sum of the static and velocity pressures; and in the second place, on having readings

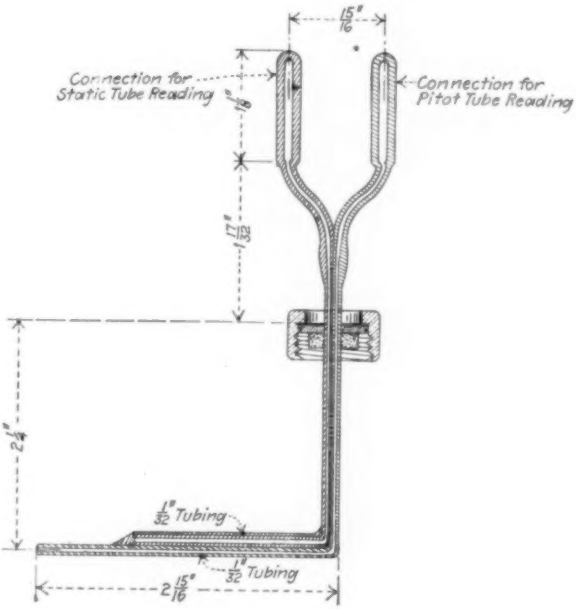


FIG. 3 DETAIL OF PITOT AND STATIC TUBES

taken at correctly established points which will give average velocity readings for the whole transmitting area.

19 No attempt was made in the design of the orifices used in the tests to make them in accordance with any mathematical formulae. These orifices served only the purpose of restricting the tool end of the hose arbitrarily, in order to give a range of readings between the vacuum behind the orifice and the volume of air handled. If vacuum and orifice or volume and orifice were to be specified instead of volume and vacuum, it would be necessary to give the details of the various orifices to be used, inasmuch as the equivalent working area and apparent area may be at wide variance.

20 The inclined manometer with gasolene for liquid was used for determining the difference between total pressure and static pressure or velocity pressure. The manometers are shown in Fig. 1. The angle of inclination for multiplying the reading was taken to suit the velocity pressure developed in order to get as long a reading and as little a percentage of error as possible. Connections to the manometer were reversed and the average taken for each reading so as further to reduce possible errors. The specific gravity of the oil was carefully determined in order that the true value of the velocity head for each reading could be computed. The density of the particular air handled was computed from government tables, taking into account barometric pressure, temperature and humidity, and from these data, for the average velocity readings, actual velocities were then computed.

21 The average velocity in feet per minute multiplied by the cross-sectional area of the 2-in. rectifying tube, expressed in square feet, gave the volume of air handled per minute for each reading. The rectifying barrel was fitted with a thermometer for taking the temperature of the air passing the pitot tube. Ordinary mercury manometers were used for the readings of the vacua behind the orifices, although in some instances standard vacuum gages were connected up for additional readings. The sling psychrometer was used for humidity and temperature determinations. Weather bureau readings were taken for the barometric pressures.

22 Applying the formula $v = \sqrt{2gh}$ to find the volume of air handled, the following equation is deduced

$$CFM = 60A\sqrt{2g\frac{W}{w}\cdot\frac{1}{12}\cdot R\sin\theta}\ldots\ldots\ldots\ldots[1]$$

where

CFM = cubic feet of free air handled per minute

A = area of rectifying barrel in sq. ft.

g = acceleration due to gravity = 32.2

W = weight of 1 cu. ft. of oil, such as used in manometer

w = weight of 1 cu. ft. of air handled, corrected for barometer, temperature and humidity

R = reading of inclined manometer in in.

θ = angle of inclination of manometer from the horizontal position

23 For the sake of convenience in reducing the apparent reading of the manometer to the true reading, the manometer supporting plate was graduated so that the manometer could be easily set to positions where its scale reading multiplied the vertical reading by 2, 5, 10, and 20, as desired, according to the particular setting used.

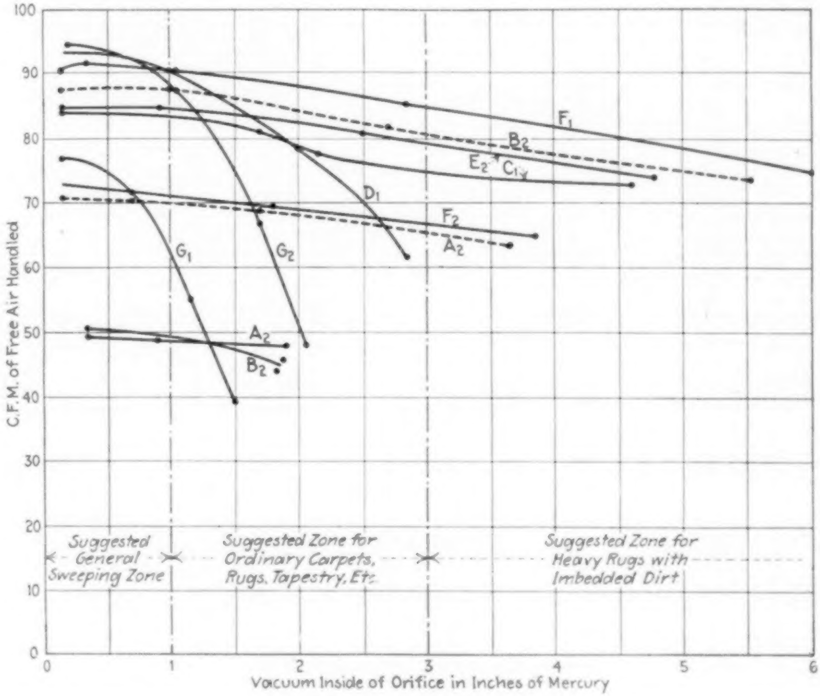


FIG. 4 CURVES OF VOLUME AND VACUUM AT TOOL FOR PIPING AND HOSE AS PROPOSED

24 In a given test, all the quantities in equation [1] can be taken constant, with the exception of R, which alone can be left under the radical, thus simplifying the reduction work for the various readings to a minimum. A scale reading directly in cubic feet of air per minute has been used by the writer as sufficiently accurate for commercial tests.

25 The results of the tests covering the ability to do work, power consumption, etc., are given in Table 1. This table gives the designating letter, type of machine, sweeper capacity of each

machine, size of piping and hose used, current consumption and other necessary data including the combined scores made up as outlined. Some of the data given in this table, and other results of the tests, are given in other form by curves, Figs. 4 to 7. Where machines did not handle per hose 80 cu. ft. of free air per minute with a vacuum of 1 in. inside the orifice, the values in columns 8, 10 and 11 are indeterminable.

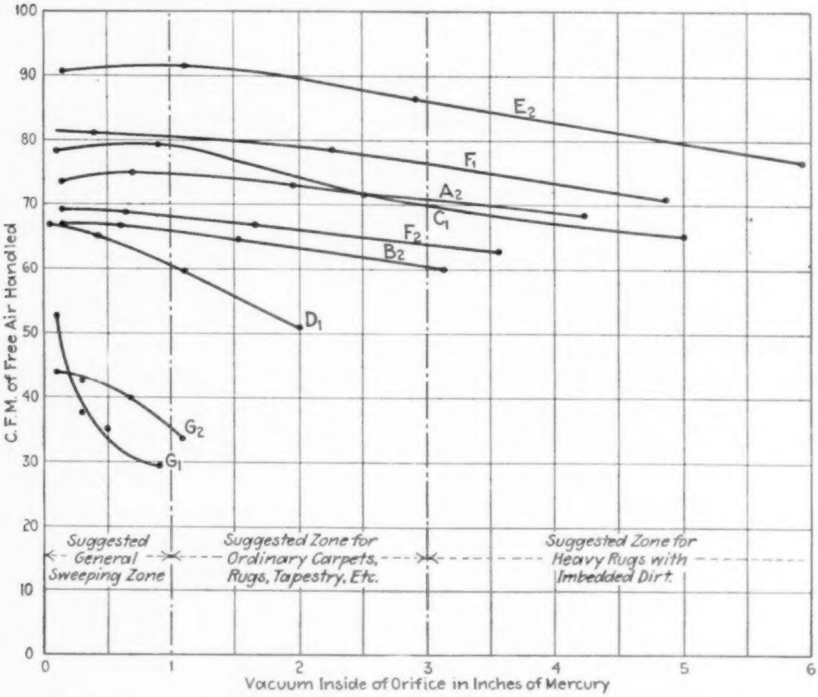


FIG. 5 CURVES OF VOLUME AND VACUUM AT TOOL FOR 3-IN. PIPING AND 1¼-IN. HOSE; THE SAME FOR ALL MACHINES

26 Fig. 4 is a series of interesting curves, plotted between cubic feet of free air handled per minute and vacuum inside the orifice at the tool end of the hose for the various machines, operating with the piping and hose system as proposed or required by the manufacturer and designated as Series *A*. The types of machines and sweeper capacities, together with sizes of hose and piping, can be obtained from Table 1.

27 Fig. 5 gives an interesting series of curves plotted between cubic feet of free air handled and vacuum behind the orifice for

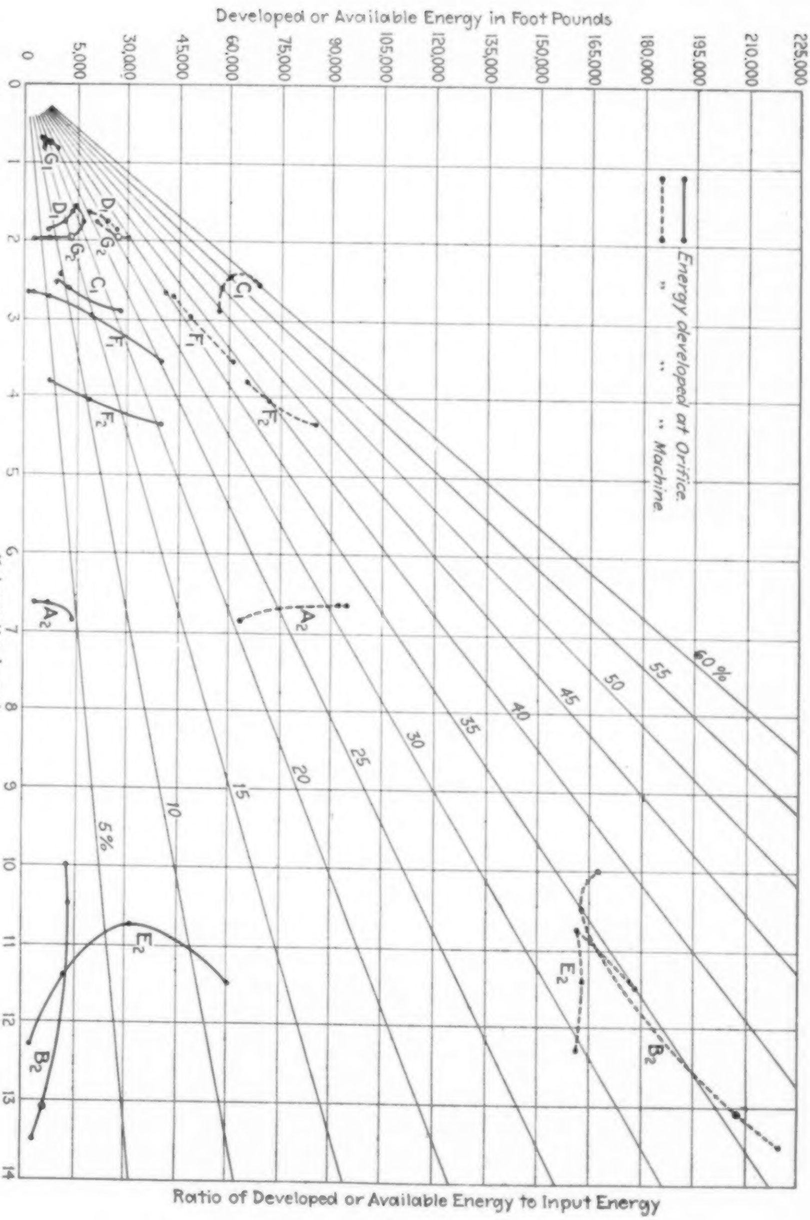


Fig. 6 Curves, with Efficiency Lines, for Energy Developed at Machine and at Tool with Piping and Hose as Proposed

the various machines operating on the 3-in. piping and 1¼-in. hose; the same for all machines outlined above as Series *B* of the tests.

28 In Figs. 4 and 5 have been drawn zone lines as boundary vacuum lines for effective work in the classes indicated. These are suggestive only, and doubtless would be placed differently by different investigators. It is probable, when vacuum cleaner tests are standardized, that some similar zone boundaries will be established and the qualifications of various machines for different classes of work be thus compared.

29 Without trying to determine whether the expansion of the air through the working orifice or through the piping system is isothermal, adiabatic or something else, the available energy or work developed at the orifice and at the machine end of the piping system for all practical purposes can be expressed in foot-pounds per minute, by the product of the rarified volume per minute and vacuum in pounds per square foot. To express it algebraicly

$$E_1 = 144 \times 0.49\ (p - p_1)\ v\frac{p}{p_1} = 70.704 m_1 v\frac{p}{p_1} = 70.704 m_1 v_1 . \quad [2]$$

$$E_2 = 144 \times 0.49\ (p - p_2)\ v\frac{p}{p_2} = 70.704 m_2 v\frac{p}{p_2} = 70.704 m_2 v_2 . . \quad [3]$$

where

E_1 = energy inside of orifice, in ft-lb. per min.
E_2 = energy at machine in ft-lb. per min.
p = barometric pressure of free air in in. of mercury
p_1 = absolute pressure inside of orifice in in. of mercury
p_2 = absolute pressure inside piping system at machine end in in. of mercury
v = volume of free air handled per min. in cu. ft.
v_1 = corresponding volume of air inside of orifice
v_2 = corresponding volume of air at machine
m_1 = vacuum inside of orifice in in. of mercury
m_2 = vacuum at machine in in. of mercury

30 Fig. 6 shows curves for the various machines plotted between the electrical input and the developed or available energy in ft-lb. at the orifice and at the machine end of the piping system for the machines connected to the particular hose and piping system proposed by the manufacturer, as outlined above for Series *A*. Fig. 7 gives similar curves for the tests of Series *B*.

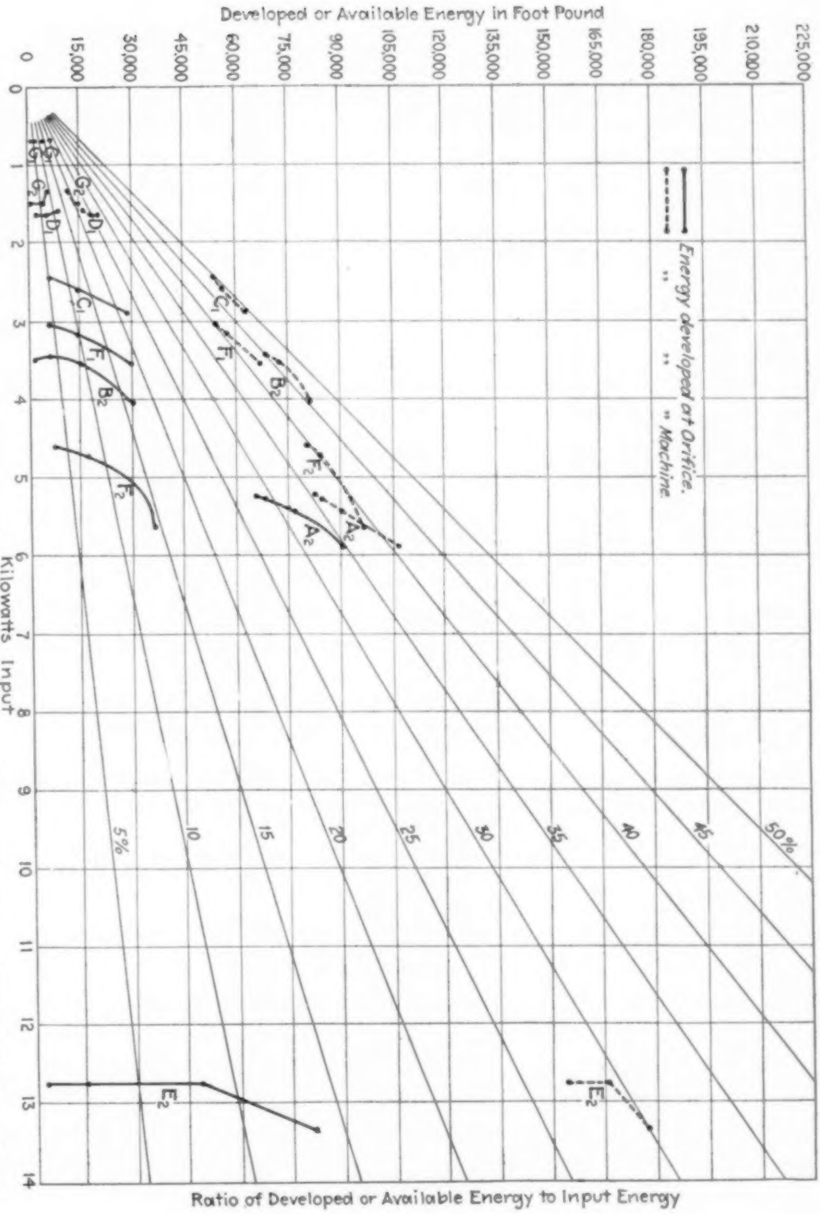


FIG. 7 CURVES, WITH EFFICIENCY LINES, FOR ENERGY DEVELOPED AT MACHINE AND AT TOOL WITH 3-IN. PIPING AND 1¼-IN. HOSE; THE SAME FOR ALL MACHINES

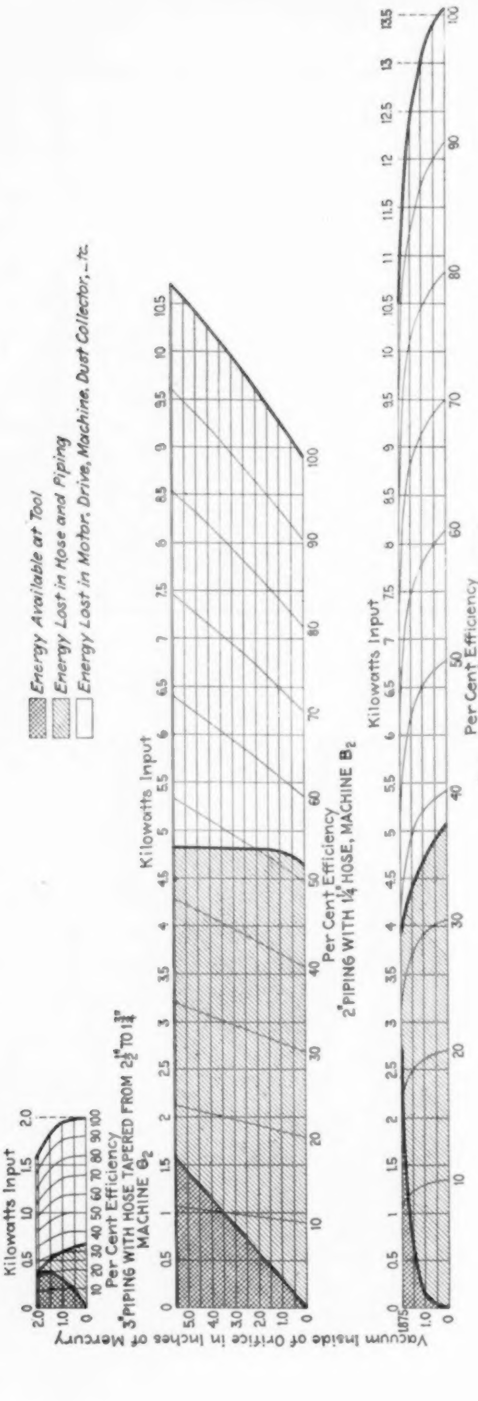


FIG. 8 GRAPHICAL REPRESENTATION OF ENERGY AVAILABLE AT TOOL AND LOSSES IN HOSE, PIPING AND MACHINE FOR A HIGH-VACUUM AND A LOW-VACUUM MACHINE

The full lines are for the energy developed at the orifice and the dotted lines for that at the machine.

31 In explanation of Figs. 6 and 7, take curve C_1 in dotted line in Fig 6. For the point of maximum input there is approximately 2.9 kw. or 128,800 equivalent ft-lb. For this input, this vacuum cleaner, which was the single-sweeper, rotary-exhauster type, gave at the machine end of the piping system, 56,550 ft-lb. of energy, or approximately 44 per cent of the total input. For the corresponding input of 2.9 kw., this machine developed at the end of the hose approximately 28,080 ft-lb. of energy, as shown by the corresponding curve in heavy line. This was approximately 22 per cent of the total input, or half the energy available at the machine end of the piping system and represents the condition of greatest restriction of the end of the hose. By similar comparisons, Figs. 6 and 7 give interesting data covering the range of efficiency in production of energy at the tool and at the machine for the various outfits tested. The ratio between the developed or available energy and the input energy is shown by the radiating lines. It is interesting to note that for two-sweeper machines the input ranged from less than 2 kw. to over 13 kw. for machines with piping and hose as proposed to accomplish the same specified results at the tool. The great difference in power consumption was mainly due to the difference in sizes of piping and hose, particularly the latter. To make this more conspicuous the two-sweeper machines having the lowest and highest power consumption as shown by the curves G_2 and the curves B_2 have been chosen. On these curves are marked conspicuously in larger circles the points coinciding with 1-in. vacuum inside the orifice.

32 Fig. 8 gives for the tests of Series *A*, graphical comparisons of these two machines as regards the power to drive, the losses through machine, piping and hose system and the energy available at the tool end of the hose. The upper diagram is for a single-stage fan machine operating in a 3-in. system and two 75-ft. lengths of hose tapering from 2½ in. at the piping inlet down to 1¾ in. at the tool. The lower diagram is for a rotary exhauster type of machine, operating on a 2-in. piping system with two 75-ft. lengths of 1-in. hose, which was the size submitted by the manufacturers for the tests. The middle diagram gives the results for the latter machine when operating on the same piping system but with two 75-ft. lengths of 1¼-in. hose, which was the smallest size permitted by the specifications.

33 These diagrams are made for power consumption losses and available energy under the full range of test conditions. Selecting, for example, the specification requirements of 1-in. vacuum inside the orifice, it is interesting to note that the machine of the upper diagram handled approximately 89 cu. ft. of free air per sweeper; the machine of the lower diagram 49 cu. ft. of free air per sweeper with the 1-in. hose and approximately 88 cu. ft. with the 1¼-in. hose, as will be seen by Fig. 4. The vacuum at the machine for G_2 to accomplish this work was only 2 in. of mercury while with B_2, with the 1-in. hose, it was 14¾ in. of mercury and with the 1¼-in. hose 10.8 in. of mercury. It is interesting to note for the 1-in. vacuum inside of the orifice, that with G_2 less than 2 kw. were required to drive the outfit and of this energy 68.8 per cent was absorbed in the machine and dust collector, 16.2 per cent in the piping system, and 15 per cent was available at the tool. For B_2 using the 1-in. hose, over 13-kw. input was required and of this 65¼ per cent was absorbed in the machine, dust collector, etc., 34.51 per cent in the piping and hose system, giving only 1.24 per cent available energy at the tool. The marked increase shown by the middle figure for B_2, using the 1¼-in. hose with other conditions the same, is interesting. There is a total input of 9.25 kw. of which 48.27 per cent is consumed by the machine, dust collector, etc., 48.58 per cent by the piping and hose system, and 3.15 per cent is available at the tool. Changing the two 75-ft. lengths of hose from 1 in. to 1¼ in. decreased the input requirements approximately 30 per cent and increased the available energy at the tool 150 per cent. Similar comparisons can be made for other vacua inside the orifice. Corresponding volumes can be obtained from Fig. 4.

34 Fig. 8 pictures some facts that will be startling to those who have not thought about the marked saving in power and the increased ability to do work that result from the use of larger hose and piping. The clear parts of the diagrams in this figure represent energy absorbed by the machine, including motor and dust collector losses, etc. The hatched portions of the diagrams represent losses in hose and piping. The double-hatched parts of the diagrams represent energy available at the tool. The important query is: What would the machine B_2 do if it were put on the piping and hose system of G_2?

The tests to determine the ability to do work, including power consumption, were conducted by Prof. H. C. Anderson and the writer. For a care-

ful analysis of the machines as machines, the author is indebted to a committee consisting of Prof. John R. Allen, of the University of Michigan; Mr. Charles H. Treat, chief designer of the American Blower Company, and Mr. Howard E. Coffin, vice-president and chief engineer of the Hudson Motor Car Company. In the acceptance tests which came a little later, he is indebted to Prof. E. J. Fermier, of the Agricultural and Mechanical College of Texas. All of these gentlemen are members of the Society.

STANDARD INVOLUTE GEARING

MAJORITY REPORT OF THE COMMITTEE ON STANDARDS FOR INVOLUTE GEARS

An outline of the work contemplated by the Committee on Involute Gears was presented to the Society at its joint meeting with the Institution of Mechanical Engineers in England three years ago, and since then the experiments on friction losses in gear teeth have been continued at the Massachusetts Institute of Technology by H. S. Waite, under the supervision of Professor Lanza, and in Philadelphia by Everett St. John at the writer's plant under his own direction.

2 The roller bearings originally provided in the apparatus for measuring the friction losses in the transmission of power were found to be the cause of serious variations in the results obtained at successive trials. The rollers developed a tendency to travel longitudinally and cause at times excessive friction by end thrust against their retaining rings, while at other times they ran freely, and thus it became impossible to distinguish accurately between the friction in the bearings and the friction in the teeth. Another difficulty developed in the alignment of the driving shaft, upon the accuracy of which the results were found to depend to a very great extent, and to overcome these difficulties, the testing machine, as previously described, was remodeled. Ball bearings were substituted throughout for the roller bearings originally employed, and one of the supporting knife edges was discarded and in its place was substituted a pair of annular ball-bearings surrounding the driving shaft to act as a fulcrum for the measurement of the driving torque. These changes entailed some sacrifice in convenience of adjustment and manipulation, but they were attended by compensating advantages in the accuracy of the results obtained, and it may now be confidently asserted that under ordinary working conditions the friction loss between the teeth of cut gears and pinions seldom exceeds one or

Presented at the Spring Meeting, Baltimore, 1913, of The American Society of Mechanical Engineers, 29 West 39th Street, New York. Subject to revision.

two per cent of the power transmitted and is practically independent of the obliquity of action.

3 The length of the addendum exerts a much greater influence than the obliquity of action, as shown by experiments made on a pinion having nominally 15 deg. obliquity, but an unusually long addendum. These gears were among those cut by Mr. Bilgram to demonstrate the practicability of making involute gears of moderate obliquity to run with small pinions and with each other. Here the addendum is prolonged as shown in Fig. 6 of the previous report,[1] and, as would naturally be expected from a theoretical analysis, the friction loss is considerably increased, the readings taken running from 1.6 per cent to 2.8 per cent as against 1 per cent to 2.2 per cent for the other types. These special gears, however, are not uniform in their proportions and strictly speaking are not interchangeable in the common acceptation of that term, because the center distance between mating gears is not constant for different combinations having the same total number of teeth. At the same time it is interesting to note the influence of the addendum as the dominating cause of friction in involute gearing, regardless of obliquity. Therefore, as far as friction is concerned, there can no longer be any doubt that the obliquity may be increased as far as necessary to avoid interference between a rack and the smallest pinion with which it is expected to engage.

4 By common consent, or, as the outcome of good practice, pinions with less than 12 teeth are seldom found in first-class machinery, and for an interchangeable involute system of external gearing, extending between the limits of a rack and any assumed pinion, it has been shown by Mr. Flanders in his admirable paper which led to the appointment of a committee on gearing, that the addendum adopted fixes the minimum obliquity to avoid interference between two limiting pinions, and that the obliquity adopted fixes the maximum addendum between the limiting pinion and a rack. Therefore, having given a 12-toothed pinion with standard addendum equal to one module, it follows that the obliquity of the system in which 12 is the smallest number of teeth must be 24 deg., and to reduce this obliquity, we must either reduce the addendum or increase the minimum number of teeth in the system. But since 12 has been generally adopted as the minimum number of teeth in all well established systems, a reduction in obliquity must be sought in a reduced addendum, and coming down to $22\frac{1}{2}$ deg., or one-

[1] Trans. Am. Soc. M. E., vol. 32, p. 823.

quarter of a right angle, it appears that the addendum cannot exceed $\frac{7}{8}$ module or about 0.28 pitch.

5 In such a system the arc of action varies from 1.23 for two 12-toothed pinions to 1.39 for a rack and pinion. Large gears running together may have an arc of action as high as 1.5, but such combinations as generally occur in practice do not cover a greater arc of action than a 12-toothed pinion and rack. Coming down again to 20 deg. obliquity we find that the addendum can not exceed 0.7 module or about 0.22 pitch to avoid interference between a rack and a pinion of 12 teeth, but with gears of 80 teeth or less, it is possible to make the addendum 0.75 module or 0.24 pitch as is done in the 20-deg. stub-tooth. Since gears of more than 80 teeth are unusual, these proportions cause no interference in the great majority of cases, or so little as to be practically imperceptible. In this system the arc of action varies from 1.12 for two 12-toothed pinions to 1.25 for a rack and pinion, and the average for various combinations is not far from the latter figure. A great many gears of this type are in service and give excellent satisfaction, and, as will be seen from the tabulated results of experiment, they consume somewhat less power in friction than the other types tested, with the exception of the B. & S. Standard, which now appears to have the lead, although in the previous experiments a difference was found in favor of the stub-tooth. These apparent variations in friction may be due to errors in readings taken with roller bearings, and although the B. & S. Standard has a longer addendum than either the stub-tooth or the proposed new standard, its active or effective addendum may really be less in new gears by reason of the modified forms used to avoid interference. However this may be, contact must sooner or later be extended over the whole addendum as the natural result of wear, at which time the greatest loss in friction would be anticipated in teeth with the greatest length of addendum. The differences however are slight and hardly worth consideration in the selection of a standard. But, in reducing the obliquity from $22\frac{1}{2}$ deg. to 20 deg. we have also reduced the arc of action, and coming down again to $14\frac{1}{2}$ deg. obliquity we find that the addendum for a 12-toothed pinion and a rack can not exceed 0.38 module or about 0.12 pitch, and that the arc of action becomes actually less than the pitch. Such a system is, therefore, impossible, and the so-called $14\frac{1}{2}$-deg. involute system, which is better known as the B. & S. Standard, necessarily includes other curves than involutes. In this system the addendum is one module and the arc of action varies

from 1.35 for two pinions of 12 teeth to 1.47 for a rack and pinion with an average for various combinations of about the same amount. This long arc of action contributes, of course, to smoothness in running when the center distances are right, but the compound curves used in forming the teeth do not admit of the variation in center distance which is known as one of the chief advantages in the involute over all other forms of gearing.

6 It is well known that all forms of gear teeth tend to wear out of shape and that the distribution of wear on involutes is more unequal than on cycloidal forms, but the unequal wear naturally relieves the pressure on the points of involute teeth more than upon those of other forms and causes a redistribution of pressure calculated to favor smoothness and quietness in running. To some extent this is also true of the $14\frac{1}{2}$-deg. modified involute standard, the gears of which furnished by the Brown & Sharpe Mfg. Co. were remarkably well made and ran under certain conditions with comparatively little noise, but whether the quiet running of these gears should be ascribed to some peculiar merit in the forms of the teeth or simply to superior accuracy in forming and spacing is a question not yet clearly determined. In regard to this Mr. Waite remarks as follows:

> The very smooth sound of the B. & S. gear as compared to the others was especially noticeable at low speeds and light loads, where the difference in the quality of the sound as well as quantity was very markedly in favor of the B. & S. gear. I believe this was largely due to the fine finish of the tooth surfaces.

7 But, of course, it may have been due to superior workmanship in forming and spacing, or as claimed by some advocates of this standard to a peculiar merit in the modified forms of the teeth which are involutes near the pitch line and more or less cycloidal in the remote portions of the faces and flanks. If the flanks of the 12-toothed pinion are radial, as they appear to be, the modification of the points of the teeth of the mating rack is necessarily cycloidal, but no exact information has been given and no effort has been made by the committee to discover the modifications used, because the key to any system of gearing is known to lie in the form adopted for the rack tooth, and when this is used as a generator, all other forms in the system naturally follow as a matter of course.

8 A good deal has been said and written about the advantages of the B. & S. Standard which cannot be denied, and it is so well established and so generally liked by its users that it will undoubtedly continue in favor regardless of objections such as may be raised against it from various points of view. The cycloidal system which

preceded it was also a good one and for many years it had no serious rival, but neither of these is an involute system and cannot claim the advantages inherent in the involute form of tooth.

9 The committee was appointed to investigate the subject of interchangeable involute gearing, and if found desirable, to recommend a standard or standards. Some of its members have favored the B. & S. Standard or none at all, while others could not admit the necessity for any modification of involute forms and preferred a solution which would avoid interference by the use of such obliquity and addendum as would reach the desired end.

10 A majority of the committee believes that a standard system of involute gearing should include in its scope a 12-toothed pinion, and all combinations between that and a rack, and that the most desirable system from every point of view is that which insures absolute freedom from interference between all combinations, with a liberal arc of action abundantly to cover the pitch in every case. To meet these conditions, nothing better has been found than an obliquity of $22\frac{1}{2}$ deg. and an addendum of $\frac{7}{8}$ module or 0.28 pitch, to which should be added for completeness a dedendum of one module, and fillets generated by a rack tooth with rounded corners prolonged to one module.

11 Such a system has very decided advantages against which must be weighed the objections raised as to journal pressure, back lash and wear. Although the thrust between centers may be 50 per cent greater than for the B. & S. Standard, it has been shown repeatedly that this additional thrust, when compounded as it must be with the tangential driving force, adds less than 5 per cent to the actual journal pressure, and it is therefore not a serious consideration. In regard to back lash, it may be said that the same care in cutting should produce equally accurate results, and that although the same depth of wear will certainly cause back lash to increase with obliquity, compensation for this appears in the fact that the greater the obliquity the better the distribution of wear. This combination of obliquity and addendum also makes very strong teeth and the obliquity adds materially to the facility with which they can be milled or hobbed, and ground when need be, after hardening. Special considerations will no doubt always influence the choice of these two variables, obliquity and addendum, and in some classes of machinery where the pinions may never have less than 15 or 18 teeth, less obliquity might reasonably be employed, but for all kinds of practice in general, a majority of the committee favors the adoption of

SUMMARY OF RESULTS

BROWN & SHARPE 14 1/2 DEG. ADDENDUM—1 MODULE

Tooth Load, Lb.	325 r. p. m.			535 r. p. m.			870 r. p. m.		
	Total Friction	Bearing Friction	Tooth Friction	Total Friction	Bearing Friction	Tooth Friction	Total Friction	Bearing Friction	Tooth Friction
1000	0.018	0.001	0.017	0.017	0.001	0.016	0.015	0.001	0.014
2000	0.014	0.001	0.013	0.013	0.001	0.012	0.011	0.001	0.010
3000	0.014	0.001	0.013	0.013	0.001	0.012	0.011	0.001	0.010

20 DEG. INVOLUTE STUB-TOOTH. ADDENDUM—3/4 MODULE APPROXIMATELY

Tooth Load, Lb.	325 r. p. m.			535 r. p. m.			870 r. p. m.		
	Total Friction	Bearing Friction	Tooth Friction	Total Friction	Bearing Friction	Tooth Friction	Total Friction	Bearing Friction	Tooth Friction
1000	0.022	0.001	0.021	0.023	0.001	0.022	0.018	0.001	0.017
2000	0.016	0.001	0.015	0.015	0.001	0.014	0.012	0.001	0.011
3000	0.018	0.001	0.017	0.016	0.001	0.015	0.013	0.001	0.012

PROPOSED 22 1/2 DEG. STANDARD. ADDENDUM—7/8 MODULE

Tooth Load, Lb.	325 r. p. m.			535 r. p. m.			870 r. p. m.		
	Total Friction	Bearing Friction	Tooth Friction	Total Friction	Bearing Friction	Tooth Friction	Total Friction	Bearing Friction	Tooth Friction
1000	0.020	0.001	0.019	0.022	0.001	0.021	0.021	0.001	0.020
2000	0.014	0.001	0.013	0.013	0.001	0.012	0.013	0.001	0.012
3000	0.013	0.001	0.012	0.013	0.001	0.012	0.013	0.001	0.012

BILGRAM 15 DEG. ADDENDUM VARIABLE

Tooth Load, Lb.	325 r. p. m.			535 r. p. m.			870 r. p. m.		
	Total Friction	Bearing Friction	Tooth Friction	Total Friction	Bearing Friction	Tooth Friction	Total Friction	Bearing Friction	Tooth Friction
1000	0.029	0.001	0.028	0.029	0.001	0.028	0.023	0.001	0.022
2000	0.025	0.001	0.024	0.021	0.001	0.020	0.017	0.001	0.016
3000	0.024	0.001	0.023	0.022	0.001	0.021	0.019	0.001	0.018

In these experiments, the teeth were lubricated with cylinder oil, the greater part of which was thrown off by centrifugal force, leaving the surfaces in a condition which might be described as slightly greasy.

$22\frac{1}{2}$ deg. obliquity with an addendum of $\frac{7}{8}$ module. That such teeth will run smoothly and efficiently has been clearly demonstrated, but the experiments have not been sufficiently prolonged to determine the effects of wear and the resulting noise. A good deal has been accomplished, however, as shown by the summary of the results obtained by Mr. St. John. These are substantially the same as those obtained by Mr. Waite, but differ slightly in some particulars, easily explained by the difficulties in construction under which Mr. Waite labored and the improved apparatus used by Mr. St. John. Journal friction on the ball-bearings used in these experiments is almost a negligible quantity and the figures given for friction on the teeth are so consistently confirmed by repeated trials that there can be but little if any doubt of their truth.

12 As pointed out by Mr. Gabriel,[1] the field for experimental research is so large that the coöperation of all interested should be secured and a fund be raised for the purpose of employing competent engineers to conduct exhaustive tests with suitable apparatus, and in this opinion the writer is in hearty accord after attempting in a small way to carry on a series of experiments upon one phase only of the problem. But as now constituted and without substantial resources at its command, the Committee on Standards for Involute Gears has but little time or means for experimental research and it has probably accomplished all that can reasonably be expected of it. Its members differ in their view points so widely that but little hope of a unanimous report has been entertained and the above is presented simply as the report of the majority, leaving the way open for dissenting members to file separate reports if they choose to do so. Under the circumstances the writer suggests that the work of the committee be considered as finished and the committee discharged.

WILFRED LEWIS, *Chairman* GAETANO LANZA HUGO BILGRAM	*Members Committee on Standards for Involute Gears*

No report has been prepared by E. R. Fellows and C. R. Gabriel, constituting the minority.—EDITOR.

DISCUSSION

LUTHER D. BURLINGAME. This question as to what is the best system of gearing for general use, a question which has been in the air for a number of years, is now sharpened by the recom-

[1] Trans. Am. Soc. M. E., vol. 32, p. 838.

mendation by the committee of a definite plan. This plan while frankly admitted by the committee as not giving the best results as to quiet running and efficiency, is nevertheless recommended by them largely because of the theoretical advantage of having the teeth a true involute throughout their entire length, thus taking a position which would seem to be giving up substance to secure shadow, a recommendation to abandon a system having not only the advantages of quiet running and efficiency, but also the further advantage of being long established and extensively used, and to adopt one which is new, and which would involve great expense in tools, cutters, etc., in changing over and would lead to the confusion naturally attending the addition of another system to that already so firmly established as not to readily be replaced.

It has been pointed out repeatedly in the previous discussion of this subject that in practice the best results are not obtained by making the teeth truly involute their entire length, this being true even when no modification is necessary in order to include a 12-tooth pinion; but rather that the points of the teeth should in any case be eased off to an extent such as experience has shown will give the best results for quiet running. Shortening the teeth does not accomplish this desired result unless the points of the shorter teeth are also eased off, thus still further shortening the working surfaces.

P. V. Vernon, representing a prominent British manufacturer of machine tools, discussing the earlier presentation of the investigations of this committee by Mr. Lewis at the joint meeting in England three years ago,[1] said that his firm had made long and costly experiments with a view of obtaining a system of gears that would run silently at high speed. The results of the experiments which his own firm and others had made had proved that there was one kind of gearing which was quieter than all others, namely the gear with "faked" teeth. It was well known that gears which gave the quietest results in running were those on which an empirical correction of tooth form had been made (referring to the B. & S. Standard).

This opinion is fully corroborated by the tests made under the direction of this committee. In Par. 5 it is admitted that the B. & S. Standard gears run with less friction and more

[1] Trans. Am. Soc. M. E., vol. 32, p. 837

quietly than gears of any other system tested. The report, however, refers to previous experiments as showing other systems to give better results, in spite of the fact that the committee condemned these previous experiments because the roller bearings used in them "were found to be the cause of serious variations in the results obtained at successive trials." On the other hand the improved apparatus, which demonstrated that the B. & S. Standard gears run more quietly and with less friction than any of the other systems of gearing tested, has been so perfected that while the "changes entailed some sacrifice in convenience they were attended by compensating advantages in the accuracy of the results obtained."

It does not seem to come with good grace from the members of the committee signing this report to minimize the importance of the results now obtained because these results are opposite to what they would wish in order to support their theory; or for them to throw the matter off with the statement that "the differences, however, are slight and hardly worth consideration." After the committee have gone to such trouble and expense as to test these very features of friction and noise and have emphasized their importance in the past, as where Mr. Lewis in his previous report[1] said that these experiments should "throw a flood of light on the problem in hand," it seems rather late in the day to say that it is "hardly worth consideration," even if the differences were slight. If I understand the tabulated results the differences are not slight. Taking all the speeds and tooth loads tabulated, the B. & S. Standard is shown to run with an average of 12 per cent less tooth friction than the best of the other systems tested. For the highest speed the B. & S. Standard shows 22 per cent less tooth friction. Assuming the same percentage of gain in durability and quiet running the difference cannot be considered as slight and hardly worth consideration.

The statement in Par. 6 that the 14½-deg. gears showed superior running qualities probably because of better workmanship, would hardly seem a plausible one as all of the gears under test were cut by experts. Such a statement looks like an attempt to explain away facts to help a theory.

I will venture the assertion that the difference in favor of the 14½-deg. gears will show even more favorably as compared with

[1] Trans. Am. Soc. M. E., vol. 32, p. 823.

the others, after having been run to a point where all the sets show wear; also that they would show up to greater advantage if all of the sets were run with a given degree of error in alignment, that is, assuming that the other gears are cut with a true involute curve the entire length of the tooth.

The report is misleading when it says of the gears with 14½-deg. pressure angle that "they do not admit of the variations in center distance which is known as one of the chief advantages in the involute over all other forms of gearing." That they do in practice allow of such variation can be demonstrated by the gears of this system now in the possession of the committee, and the favorable comparison in this respect with other systems, will, it is believed, be even more evident as the gears become worn, the easing off of the points of the teeth being such as to reduce the wear and minimize its evil effects. This additional length of tooth thus serves a useful purpose and is believed to be one of the chief reasons for the quiet running of these gears as shown in the tests reported, as it gives an easy action of approach instead of having a sharp corner with its tendency to a gouging action, as in the case of a short tooth made involute its entire length.

Noise means wear, and quiet running not only means less racking of the nerves, but greater durability and financial saving. Less wear means less back-lash, as does also the smaller pressure angle where a given amount of wear produces less back-lash than in the case of the greater pressure angle.

Such conditions of wear, inaccuracy of alignment and varying center distance, are the practical conditions under which the majority of the gearing in use is run, and any system which does not most fully provide for such needs cannot be recommended for general adoption.

I trust that it may not be felt that, as representing the Brown & Sharpe Mfg. Co., I am coming forward to oppose any change from the established standard simply for commercial reasons. If commercial reasons were the ones governing, a change might be encouraged which would throw the burden of equipping with new cutters and hobs on the manufacturing public, because this change while proving expensive for the user of cutters, would bring new business to the cutter manufacturers. It would mean to the user not only the loss and confusion due to having part of his equipment of cutters of one system and part of another, not

interchangeable with each other, but it would require more cutters in a set to maintain the same standard as at present, i. e., where eight cutters now make a set to cut from a 12-tooth pinion to a rack for ordinary work and where 15 cutters make a set for work where the conditions are more exacting, the adoption of a system with a greater pressure angle and shorter teeth would require more cutters in each of these sets, all of which would make business for the manufacturer of cutters.

My company is equipped to make cutters or hobs by the proposed system as well as by the present, but in the interests of the user and for the reputation of the maker, such cutters would not be made involute the full length of the tooth unless specifically ordered to be so made.

Long experience with the gear question and its many pitfalls, leads us to take a conservative position at this time and to give a word of caution as to any action which in the light of future experience might be found to have been hasty. We believe that no action should be taken leading away from the present established gear-tooth system until after the exhaustive experiments under actual working conditions recommended by the committee have been made, and have proved conclusively that some new system meets those working conditions better than that now in common use.

Henry Hess. The committee have done yeoman work. They have given enormously of their time and themselves carried a heavy burden of expense, and the special thanks of our members are due them.

There is one decided objection to the Brown & Sharpe system: It does not lie in the performance of those gears; it lies in the fact that no one knows what that system is. It is a system of a certain general type modified, as its sponsors say, necessarily, but aside from those manufacturers, no one person knows definitely what that modification is. It may be laid down as axiomatic that no system can hope to receive recognition as a standard which is confined to a single firm, no matter what the eminence of that firm nor how well it may have performed its work, not even though it have the deservedly high standing that B. & S. means the world over.

The Committee. In the discussion of this report, Mr. Burlingame seems to lose sight of the fact that, until the Brown &

Sharpe Standard has been mechanically defined, it is really anything that Brown & Sharpe choose to make, and, as stated by Mr. Hess, no one, aside from its sponsors, knows exactly what it is. Some criticism has also been made on the want of uniformity in the cutters furnished for the same purpose at different times, and although some variation might reasonably be expected in the output of such products, it is impossible for the user to know what is right or what is wrong in the absence of a comprehensive definition which might enable others skilled in the art to make and use such cutters.

Mr. Burlingame also loses sight of the fact that the Brown & Sharpe Standard is not preferred exclusively on account of its running qualities and that "the very smooth sound of the Brown & Sharpe gear was especially noticeable at low speeds and light loads." Quite as many, if not more correspondents, preferred the stub tooth, and a further comparison of the Brown & Sharpe gear with the sample 22½-deg. involute gear, has satisfied the majority of the committee that although the Brown & Sharpe test gears run more smoothly, that is with less vibration in the noise per revolution of the gear wheel, there is really very little difference in the average volume of sound. The 22½-deg. gear produces pulsations of sound which coincide with the revolutions of the wheel and Mr. Bilgram has no doubt that these are due to a slight eccentricity arising from cutting this gear on an overhung arbor. Unfortunately no apparatus has as yet been devised to visualize the noise and so enable definite comparisons to be made at different times. It was hoped at the outset that something of this sort might be developed at the Massachusetts Institute of Technology, but in this the committee has been disappointed and it recognizes the difficulty of making accurate comparisons between the hum of a given pair of gears while running and the hum of another pair which ran some time before. Such impressions are naturally very elusive, but it can hardly be denied that smoothness or uniformity in the noise indicates uniformity in the work and a high degree of accuracy in spacing the teeth. This admission in favor of the products of the Brown & Sharpe Manufacturing Company does not lead, however, to a conclusion in favor of the Brown & Sharpe Standard, because any standard would undoubtedly appear to the best advantage when so perfectly applied.

The committee does not doubt that for a 14½-deg. involute

system, the modifications embodied in the Brown & Sharpe Standard to avoid interference and promote interchangeability are as complete and effective as can be devised, but a majority of the committee does not recognize the necessity for so little obliquity in involute gearing as to cause any modification whatever in the true involute form. This attitude has been condemned as subservient to a theory and the contention has been made that a question of such importance should not be treated as an academic matter, but the majority of the committee does not recognize any limit to the application of sound principles nor any point in the problem to be solved beyond which such principles must give way to empirical methods. In the choice of an interchangeable system of involute gearing to include a 12-toothed pinion and a rack, it is perfectly clear in the first place that an obliquity of $14\frac{1}{2}$ deg. will not give enough arc of action between two 12-toothed pinions to cover the pitch and that when such pinions are made to run together they do not run as involute gears. If their flanks are radial as they appear to be in the B. & S. Standard, such pinions are necessarily more closely related to the discarded and condemned cycloidal system than to the involute, with the merits and defects of both only partially realized.

If the teeth do not bear to their ends when new they must do so later on as the result of wear, and if they show less friction because contact does not take place over the whole addendum when new, this is just as it should be from a theoretical standpoint. The difference, however, is slight, and if experiments were needed to prove that obliquity alone has no effect upon the friction of the teeth themselves, the evidence adduced should help to establish that fact. The close relation between friction and the working length of the addendum is also clearly indicated in the experiments and both of these relations are fully sustained by analysis.

It is admitted that a slight increase in journal pressure will result from an increase in obliquity, but it cannot be said with certainty what the obliquity of the Brown & Sharpe Standard really is, because the nominal obliquity of $14\frac{1}{2}$ deg. is all that is known, leaving the obliquity of the modifications in doubt. It is certain, however, that the increase in pressure due to an obliquity of $22\frac{1}{2}$ deg. cannot exceed 5 per cent and in all probability it is considerably less. This increase in journal pressure

has always been the chief argument against an increase in obliquity, but it does not appeal to the majority of the committee as the most important consideration in the selection of a standard system of interchangeable involute gearing.

Nothing has been said about the number of cutters in a set nor the effect of the approximations which are necessarily made in milling an unlimited number of gear sizes with a limited number of cutters. The number of cutters in a set naturally depends upon the degree of accuracy desired. Brown & Sharpe cover the field very well with 8 standard cutters, they do it better with 15 and still better for cycloidal teeth with 24 cutters; but another well known maker has never been satisfied with less than 24 cutters in a set, whether involute or cycloidal in form. To avoid pounding on the ends of teeth, the cutters selected to form a set should never be used on gears having fewer teeth than the number on which they are formed. This is the well-established practice of Brown & Sharpe and it is undoubtedly correct, but it simply mitigates the evil of using the wrong cutter.

When gears are hobbed or generated on a gear shaper, one cutter serves to make all sizes on the principle that gears which run with the same master wheel acting as a cutter, will run together. This principle is correct if the addendum of the master cutter is prolonged to cover the same arc of action as a rack cutter, and for the proposed root clearance of $1/8$ module, such master cutters should not have less than 30 teeth, or the equivalent in a hob, but of course smaller cutters can be used by increasing their addendum and consequently the dedendum of the gear cut, which is not a vital matter. The spring in the machines used also affects the results obtained, and in general it may be said that all gear teeth are more or less approximate in form whether milled or hobbed or planed. They are, therefore, all subject to improvement or depreciation, as the case may be, from wear, and this deals with quantities too minute to be measured or corrected in the cutters themselves.

Interchangeable gear teeth which are true to form and evenly spaced generally carry the whole load on one tooth at and near the pitch line, and divide it between two teeth at the remoter parts of the arc of action. In such gearing, it is very unusual to have more than two teeth engaged at one time and not at all desirable, since a large arc of action between large gears means a short

arc of action between small gears of the same series. But it is absolutely necessary to cover the pitch and to cover it well. Having done this by means of an obliquity and an addendum which avoid interference throughout the series contemplated, the relief supposed to be needed at the ends of the teeth to promote smoothness in running takes care of itself through wear. There is no need of any modifications or allowances for this purpose in involute gearing because the tendency of wear is to relieve the points and concentrate pressure in the neighborhood of the pitch line. Further than this, gears cut with cutters formed to a smaller number of teeth are necessarily relieved at their points more than the best action requires and the full arc of contact is attained only by wear.

On the other hand, cycloidal gear teeth tend to wear uniformly under constant pressure and therefore more in the neighborhood of the pitch line, where the load is undivided, than at their points where the load is divided. Consequently there is a greater tendency for such teeth to wear less at their ends and pound. It is no wonder, therefore, that the cycloidal system is "dead," as declared by Mr. Vernon in the discussion referred to by Mr. Burlingame. But if, at one time, when in favor, its demise had been predicted by an analysis of the effect of wear, as it surely might have been, the advocates of empirical methods, who know something better than theory and want to be shown in a practical way, would no doubt have stood by this standard against any reasonable arguments based upon broad general principles. But there is no difference between theory and practice when both are properly understood and applied. Each supplements and aids the other in arriving at the truth, and any sort of interchangeable gearing which is not based upon some theory of the proper action of gear teeth, is certainly not a system. It may work well enough or very well indeed as a composition, but it lacks the binding force of a unitary principle which makes the reproduction of every cutter absolutely determinate and reliable.

The proposed standard covers a wide range of numbers from 12 teeth to a rack. It gives a large increase in strength over the Brown & Sharpe Standard and a better distribution of wear. The wide angle enhances the freedom in cutting by any process, and the involute form without alteration or correction of any kind simplifies the problem of grinding after hardening.

Although no action has been taken to establish the number of

cutters in a set nor the lateral clearance between cut teeth, it is evident that the same number of cutters will approach perfection in the proposed system as well as in the Brown & Sharpe Standard, and since contact on both sides of the teeth is impossible in any approximation, it is suggested that the thickness of all cutters at the pitch line, measured on the chord, be taken at one-half the pitch measured on the pitch circle. The clearance would then be a maximum for two engaging 12-toothed pinions and in this case it would be barely 0.003 in. for 1 in. pitch, or in general not more than 0.003 of the pitch, whatever that might be. This would reduce the clearance almost to the vanishing point and yet leave enough to relieve unavoidable errors in forming and spacing. Of course, clearance or backlash will always increase with wear, but ordinarily when transmitting power there is no rattle or noise from this cause. In cases where all backlash must be eliminated and the teeth made to touch on both sides while running, the completion of the gears by grinding on each other puts them in a special class which can hardly be considered interchangeable. They may wear in this way out of their original shape but not out of rolling contact which covers an infinite variety of shapes beside the Brown & Sharpe Standard.

So, while admitting that the Brown & Sharpe Manufacturing Company has probably made the best possible adaptation of 14½-deg. obliquity to practical use, the majority of the committee cannot, for the reasons stated, accept this composition as the ideal form of gearing which should be adopted for an interchangeable involute standard.

ECONOMICS OF CENTRAL STATION HEATING

By Byron T. Gifford, Chicago, Ill.

Junior Member of the Society

The first meeting of the newly organized section of the Society in Chicago was held on April 9, 1913. About 55 members and guests were in attendance and Paul M. Chamberlain, chairman of the meeting, presided. The question of holding meetings jointly with the Western Society of Engineers was discussed, and it was decided that such an arrangement would not be attempted at present, although it might be brought about at a later time. It was further voted that in addition to the professional meetings planned for the coming year, one meeting devoted almost entirely to social matters should be held in order that the members might become better acquainted.

Byron T. Gifford, manager of the engineering department of the American District Steam Company, then read a paper on the Economics of Central Station Heating, an abstract of which follows.

ABSTRACT OF PAPER

There are three important branches of central station heating: (*a*) the production of heat units; (*b*) the distribution of this product; and (*c*) making the service attractive to the consumer.

The first of these, the generation of heat, may be handled by three distinct methods: by direct firing, or "straight fuel burning" plants; by an electric generating plant with heating as a by-product; and by a combination of a by-product plant and a heating plant.

Owing to many causes the heating plants of the future will be of the last type. In an arrangement of this kind, electricity, for instance, can be produced much more cheaply than is possible in the most economical electric plant, and there are a number

---

The American Society of Mechanical Engineers, 29 West 39th Street, New York.

of ways of disposing of this by-product: It may be sold to an existing electric plant at a wholesale rate; a market may be made for it by creating industries requiring a fairly large amount of electricity at a comparatively low price. The author has in mind the case of an industry which has supplied to it 400 kw-hr. of electricity 20 hours per day, 365 days per year, at less than 1 cent per kw-hr. The net income to the heating company from this by-product is more than sufficient to meet the payments of the bond interest, taxes and insurance of the entire property. This additional income, amounting approximately to $11,000, is being handled with an increased operating cost of $2200, as compared with the operation of the plant the year previous when no electric current was generated. With a simple or a twin Corliss engine a kilowatt of electricity can be generated under conditions which exist in the average central heating plant with 45 lb. of steam. Assume for example a heating plant serving 200,000 sq. ft. of steam radiation. This load gives an average demand on the boilers of about 40,000 lb. per hour, which, if sent through a simple or twin Corliss engine, will develop 880 kw-hr. of electricity. Assume that this electricity is sold to some consumer for 1 cent per kw.; at this price 1 cent would be realized for every 45 lb., or 22 cents for every 1000 lb. of steam delivered to the heating mains before it had left the station. The history of central station heating has proved that a rate averaging 60 cents per 1000 lb. for steam is reasonable and can be procured from any heating consumer. This would mean 82 cents per 1000 lb. for the steam generated, which is a good return on the investment.

Heating plants in the United States at present vary in size from a connected load of 10,000 sq. ft. of radiation to a connected load of 1,750,000 sq. ft. of radiation. These plants are built in towns of upward of 1000 people.

There are few electric generating plants that can put a kilowatt-hour of electricity on their switchboard below a cost of ½ cent per kw-hr., and there are also few electric plants that can generate and distribute to the primary side of their transformers for a cost of 1 cent per kw-hr., especially when the overhead and fixed charges are considered. Heating plants, with electricity as a by-product can do this, and this fact alone will make a place for central heating plants. Some of the larger operating com-

panies are doing this at the present time and others are building properties to operate along this plan.

Regarding the second point, distribution, in water works or in gas properties the losses from distribution consist of leaks and pressure drops due to friction which is caused by insufficient pipe capacities. In central heating there are these two losses to contend with, and in addition the loss from radiation, called "line loss" by central heating engineers. Leaks are a decided detriment to heating pipe line, and the loss from this source is probably more serious than was at first supposed. Assume for example, a hole $\frac{3}{4}$ in. in diameter in a pipe line containing steam at 5 lb. pressure. With an evaporation of 7 lb. of water per lb. of coal, this leak will cost 23 tons of coal per month. A leak of this size is equally as detrimental in a hot-water heating system. The loss from friction affects, of course, the initial pressure to be carried on the heating mains, and consequently affects the back pressure to be carried on the engines.

As to the detriment of too high a friction loss, in a 3-in. steam line 1000 ft. long, carrying 1000 lb. of steam per hour, the friction loss will be equal under average conditions approximately to 1460 B.t.u. per hour. This same load carried 1000 ft. in a $3\frac{1}{2}$-in. line will show a friction loss equal approximately to 500 B.t.u. and a 4-in. line under the same conditions will lose only 240 B.t.u. from friction.

The radiation loss from central heating is even more important than friction loss, from the B.t.u. standpoint. There is a number of styles of underground insulation in use to-day, which lose anywhere from 0.03 lb. to 0.38 lb. of steam per sq. ft. of underground surface per hour.

In order to determine the most economical line to instal it is necessary to combine the friction and radiation losses. Take, for example, a demand of 1000 lb. of steam per hour which must be carried 1000 ft. A 3-in., $3\frac{1}{2}$-in., 4-in. or $4\frac{1}{2}$-in. pipe will do this work, but the most economical size must be determined. The loss from friction would be 1460 B.t.u. per hour on the 3-in. line; 500 B.t.u. on the $3\frac{1}{2}$-in. line; 250 B.t.u. on the 4-in. line and 130 B.t.u. on the $4\frac{1}{2}$-in. line.

Assume a radiation loss of 0.05 lb. of steam per sq. ft. of underground surface per hour; the radiation loss from the 3-in. line will be approximately 45,000 B.t.u. per hour; $3\frac{1}{2}$-in. line approximately 52,000 B.t.u.; 4-in. line approximately 58,000 B.t.u.;

4½-in. line approximately 65,000 B.t.u. Combining these two losses, it is seen that the 3-in. line is the most economical to instal, provided it does not affect the heating station conditions by causing excessive back pressure on the engine.

Experience has shown that a 3-in. line carrying this load the given distance will have a drop in pressure of about 7 lb., and as it is necessary to have 1 lb. pressure at the end of the line, it means that the back pressure at the station would be approximately 8 lb. If this pressure is excessive it would be necessary to run a 3½-in. line, which would give a back pressure of less than 5 lb. The difference in cost between a 3-in. and a 3½-in. line is comparatively small. The cost of material is somewhat less for a 3-in. line; the cost of labor is approximately the same.

To show the value in dollars and cents of efficiency in underground insulation, a piece of line insulated with a construction that will lose 0.05 lb. of steam per sq. ft. of underground surface per hour may be compared with a line insulated with a construction that will lose 0.14 per lb. per sq. ft. per hour. Both constructions are found in everyday practice. For this purpose an 8-in. line 1000 ft. long will be considered. Such a pipe has 2.25 sq. ft. of surface per lineal foot. In the first case the line loss will amount to 112 lb. of steam per hour; in the second case to 215 ft. Assuming a generation cost of 30 cents per 1000 lb., in a season's operation (from October 1 to June 1, or 5832 hours) a line loss in the first case will equal $195 per year and in the second case $550 per year.

The money saved, therefore, will be the difference between these two figures, or $355 per year. This is 10 per cent on $3550, an amount which can be spent to instal the more efficient construction. Since the difference between these line losses is 0.09 lb. of steam per sq. ft. of underground surface per hour, and since the example assumed consisted of 1000 ft. of 8-in. line, which is 2250 sq. ft. of surface, it is seen that for every 0.01 lb. of steam saved on this 8-in. line, approximately $400, or 20 cents per sq. ft. of underground surface, can be spent.

Another interesting example is the comparison of two insulations on an entire underground heating installation. This installation consists of approximately 800 ft. of 12-in. pipe line, 1200 ft. of 10-in. pipe line, 3600 ft. of 8-in. pipe line, 4200 ft. of 6-in. pipe line, and 7300 ft. of 4-in. line and surface. The number of square feet of underground surface in this system is equal

approximately to 30,000 sq. ft. Comparing two insulations, one with 0.04 loss and one with 0.09 loss (difference 0.05) by the following formula which the author has derived, the more efficient insulation in this case will be worth $30,000 more than the less efficient:

$$N \times S \times L \times C$$

where

N = the difference between the two insulations in hundredths of a pound of steam lost per hour

S = the number of square feet of surface per lineal foot of pipe

L = the length of pipe

C = a constant based upon the cost of steam per thousand pounds and the number of hours in the heating season

Where the steam costs 30 cents per 1000 lb. and the heating season is 5900 hours, the value of C is 21 cents.

Regarding the third point, making the service attractive to the consumer, it is not necessary, perhaps, to wrap it up in a good looking package, but it is necessary to have it attractive in price and quality. That this class of service is attractive is shown by the fact that central heating plants, with few exceptions, have had no trouble holding their consumers. It is this one fact, as much as anything else, that has put many heating plants into trouble. They have been tempted to serve a larger territory or more consumers than their plants could economically handle, and they have been tempted also to extend their services without due regard to the economics of the proposition.

As to the best method of selling the heating service, experience has shown that the meter basis is the most equitable. The rate will depend upon local conditions. A number of companies are selling steam on the meter basis, with a sliding scale rate; others have devised a maximum demand or readiness to serve rate, both of which work out admirably in practice. The quality of the service is always sufficiently attractive to obtain a large percentage of the possible consumers.

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

CONTENTS

The Editor will be pleased to receive inquiries for further information in connection with articles reported in the Review. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

FOREIGN REVIEW

In a paper read at the Cambridge meeting of the Institution of Mechanical Engineers entitled A Few Notes on Engineering Research and Its Coördination, G. H. Roberts advocates the necessity of a definite and generally recognized system for making known the results of the numerous private researches and experiments which are continually being carried on. He makes the following statement: "The author is not unaware of the fact that many arrangements are now, and have for some time been, in operation giving references to work carried on elsewhere, and need only instance the excerpts published in the Proceedings of the Institution of Civil Engineers, of the Iron and Steel Institute, and by The American Society of Mechanical Engineers, to mention but a few. These references, excellent though they be, are, however, usually in the nature of general records of works accomplished rather than of researches carried out with definite objects in view."

It would require a more complex organization than probably exists in the case of any of the societies mentioned to establish a clearing house and advance depositary of engineering information in connection with their publications. There can be no doubt, however, that an international exchange of information concerning engineering investigation would be of great advantage and the suggestion should have consideration as indicating a direction for future development. In the Foreign Review of The American Society of Mechanical Engineers all that has been attempted has been to bring to the American engineer, in as clear a form as space permitted, such data published abroad as might be of interest to him in his day's work. The letters received at this office, as well as the fact that abstracts from The Journal are frequently reprinted in the form of regular articles by American and British technical papers, indicate that, in its own field, the Review helps the English speaking engineer to know at least what has been done abroad, even if it cannot as yet tell him of work not ready for complete publication.

THIS MONTH'S ARTICLES

The article of Hoefer on the flow processes in the ascending pipe of the Mammoth pump, though referring more particularly to this particular apparatus, applies equally well to other types of air-lift pumps, and may be of interest to hydraulic turbine engineers. Data of the Allevard tests on water hammer bring out an important fact, viz., that the water conduit pipe subject to water hammer, due to sudden closure of the outflow opening, must be made equally strong all along its length, although this is not necessary where provision is made for gradual closing only. These tests also confirm in part the Allievi theory, of which a second installment is reported in this issue. This theory is considered, and generally so recognized in Europe, of such importance that a very complete abstract, showing as far as possible the mathematical processes used by the author, is given.

Bartel indicates the economic limits of the gasification of low-grade fuels, and attempts to prove that, as matters stand now, a gas-fired boiler with a steam turbine is more efficient for large installations than a straight gas engine plant, even though the latter is more efficient thermodynamically. In the section Mechanics the first part of an article on shock absorption in power vehicles is abstracted, as well as an article on the experimental determination of the coefficient of cyclic variation by means of an eddy-current tachograph. In the same section Z. Carrière describes a convenient and certain cinematic method of measuring air velocities, particularly convenient for calibrating pitot tubes and similar instruments.

The next section contains a description of a recording load indicator showing the load fluctuations of an engine, and thus giving data interesting both in testing engines, and in many cases of their industrial operation, e.g., in isolated plants, rolling mills, etc. The Bourlet apparatus for measuring the vibrations of solid bodies in case there is no fixed support for the measuring apparatus will probably be of particular interest to automobile and aeroplane manufacturers. Data and charts in connection with blowing-off of boilers, and results of tests on resistance offered to the flow of superheated steam through smooth and corrugated expansion pipes will be found in the section Steam Engineering. In other sections will be found articles on torsion indicators, strength of soft steel bars with drilled and reamed holes, electric atomization of metals, etc.

Hydraulics

INVESTIGATION OF FLOW PROCESSES IN THE ASCENDING PIPE OF THE MAMMOTH PUMP (*Untersuchungen über die Strömungsvorgänge im Steigrohr eines Druckluftwasserhebers*, K. Hoefer. *Zeits. des Vereines deutscher Ingenieure*, vol. 57, no. 30, p. 1175, July 26, 1913. 9 pp. 17 figs., *etA*). Tests of flow processes in the so-called Mammoth pump. It was found that some data of tests do not agree with the values obtained by calculation and the author came to believe that this was due to the failure of taking into consideration the velocity of flow of the air with respect to water, when passing through the ascending pipe. It is impossible to determine this "relative velocity of air" theoretically, and the tests described have been undertaken for the main purpose of determining it experimentally. In all the tests water at ordinary temperature was used, and the ascending pipe had the same diameter of cross-section throughout its length. The tests were made in the machine construction laboratory of the Technical High School at Berlin.

The first series of tests was to determine the velocity of air bubbles rising through the water, as a function of the diameter of the bubble. Under the assumption that the bubbles are perfect spheres, the following formula ought to hold good:

$$v_L = \text{const.} \sqrt{\delta(\gamma_w - \gamma_L)} \ldots\ldots\ldots\ldots [1]$$

where δ is the diameter of the air bubble, γ_w specific weight of water, and γ_L specific weight of air. This equation shows that (*a*) at the same distance from the surface of the water, a larger bubble moves with greater velocity than a smaller one, and (*b*) the velocity of an air bubble rising through water constantly increases owing to the expansion of the air, and consequent increase of δ. Instead of following the theory as set forth in this equation, the tests have however presented the rather startling relation between diameter and speed shown in Fig. 1A, where the speed rises rapidly at first with the diameter, reaches a certain maximum value at $\delta = 1.1$ mm (0.043 in.) approximately, and begins to decrease slowly, until it reaches a minimum, from whence on it again begins to rise. This remarkable behavior of the curve may be partly explained by the fact that, contrary to the assumption of equation [1], the bubbles are not spherical; the main reason why the speed decreases is because of the fact that when the bubbles have diameters less than 1.1 mm, they rise through the water vertically, and, if deflected, resume a vertical direction again when free to do so. If, however, the diameters are even a little in excess of 1.1 mm, the path of the bubbles become helical, which explains the slower speed in the purely vertical direction. The transition from straight line to helical motion could be clearly observed in several cases. As the diameter increases still further, beyond 3 mm, the helical path of the bubble loses its regularity, and at the same time the shape of the bubbles becomes more and more unlike a sphere; the resistance of water tends to flatten them, so as to make the horizontal diameter greater than the thickness of the bubbles. The rising branch of the curve in Fig. A may be determined approximately by the equation

$$v_L = 0.275\,\delta^{1.4}\text{m/sec. for } \delta \leqq 1.1 \text{ mm} \ldots\ldots\ldots\ldots [2]$$

and

$$v_L = \frac{1}{\delta} + 0.0379\,\delta^{0.6}\text{m/sec. for } \delta \geqq 15 \text{ mm} \ldots\ldots\ldots\ldots [3]$$

For 1.1 mm < δ < 15 mm a simple mathematical relation between δ and v_L cannot be established. The above shows definitely that there is no similarity whatever between the theoretical equation [1] and the experimental equations [2] and [3]. Further experiments have shown that the speed of the motion of the bubbles through water remains the same when the pressure is somewhat in excess of the atmospheric, tests having been made with pressures up to 1.3 atmospheres gage.

A further series of tests was made to establish directly the velocity of the air bubbles in the ascending pipe of the Mammoth pump (the description of the methods used has to be omitted owing to lack of space). It has been found that, at first, the velocity in the higher parts of the ascending pipe is below that in the

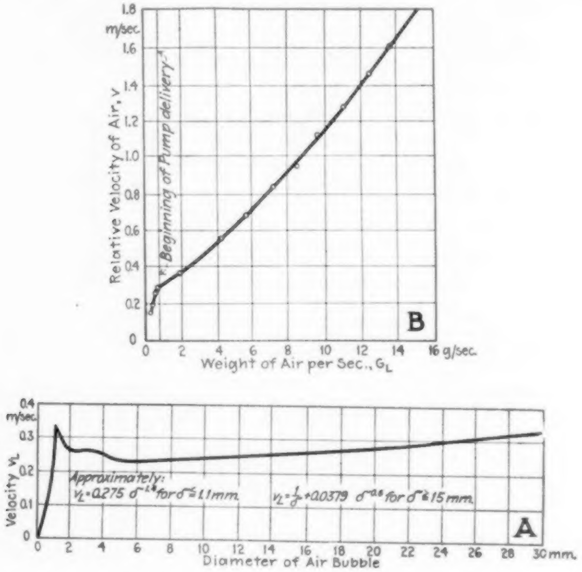


FIG. 1 FLOW OF AIR THROUGH WATER

lower strata, but increases again when the upper half of the pipe is reached. The average speed in one representative test has been found to be 0.28 mm (0.0109 in.) per sec., the air bubbles having diameters from 3 to 5 mm. Since the speed from Fig. 1A for bubbles of these diameters appears to be only 0.25 mm (0.01 in.) per sec., the speed of the smaller bubbles does not materially seem to affect the average speed of the mass of air flowing. The velocity of air increases with the increase in the amount of air flowing, as shown graphically in Fig. 1B, which shows also that the relative air velocities (relative with respect to the water) are considerably greater than was usually supposed, the increase in velocity with the increase in the amount of air flowing being due probably to the prevalence of large bubbles, and *these high values of the air velocity v are probably the cause of the low efficiency of the Mammoth pump.* The author states in conclusion that while the data reported here give an insight into the theory of the Mammoth

pump, they are still insufficient for its theoretical calculation, which would become possible only when it was known how the air velocity v is affected by the inside diameter of the ascending pipe, and by the relation between the draft and delivery head, or $E:F$. These experiments also fail to show how v may vary with still larger amounts of air flowing. The article also contains methods for calculating some of the elements of operation of Mammoth pumps, not reported here owing to lack of space.

FRENCH HYDROTECHNICAL SOCIETY (*Société Hydrotechnique de France, La Houille Blanche*, vol 12, no. 6, p. 178, June 25, 1913. 5 pp., *eg*). Subsequent to the Congress of White Coal in 1902, a French turbine commission was appointed, with an extensive program, but comparatively modest means at its disposal. It did a certain amount of useful work, but the lack of means prevented it from accomplishing all that its program promised, so that when in 1912 the French Hydrotechnical Society was formed, the commission voted to disband, and to leave to the new society the task of finishing its work. As most of the work done by the commission has been duly published, only the part referring to the action of water hammer will be reported here owing to its connection with the articles of Allievi and de-Sparre, published in this and the preceding issues of *The Journal*. Tests made at Allevard have shown that water hammer is produced by damped waves, in conformance with the theory. The first wave, absolutely flat on top, conforms fully to the views of Allievi, and by a calculation of overpressure due to the water hammer by using the formula $y - y_0 = \frac{a}{g}\Delta v$, a value is found which is in full accord with the experimental values. But, contrary to the theory of Allievi, all the following waves have a damped *sinusoidal* shape. The period, i.e. the length of waves of the water hammer, which in the Allevard experiments has been found to be equal to 1.84 seconds, also does not fully agree with the theory of Allievi, and is longer than $4L/a$, where L is the length of the conduit in meters, and a is the speed of propagation of the waves, determined from the formula

$$a = \frac{9{,}900}{\sqrt{48.3 + K\frac{d}{e}}}$$

On manometric curves certain very marked irregularities are noticeable, in the form of flattening out of the curve at certain spots, or pressure considerably below the atmospheric, or overpressures at other spots, all of these irregularities being evidently due to the influencce of a parasite wave, which is reflected when the diameter of the pipe varies, and comes back to the outflow orifice. Various curves taken from pressure records of experiments in which the flow of water was not entirely stopped, clearly show the extreme rapidity of the damping produced by a continuation of flow through the orifice, there being in that case only one or two waves, as against 20 and more in the case of complete closing.

The numerous diagrams taken in the case of sudden opening of the outlet clearly show the positive water hammer following the negative water hammer initially caused by the sudden opening. Since, however,

continuing flow very rapidly produces damping, the amplitude of the positive water hammer is notably below that of the negative. By placing recording pressure indicators at various points along the conduit, it has been found that the wave produced by a sudden closure of the conduit outflow opening travels with nearly the same intensity all along the conduit until close to its upper free opening, which is in accordance with the theory of Allievi, and is of particular importance to conduit designers as it shows that the pipe must be of such strength throughout as to be able to support an overpressure not only in its lower part, but practically up to the top free outlet. This, however, applies only to sudden overpressures, because a gradual closing produces a variation which in its propagation meets with the interference due to other waves coming in the opposite direction, or produced by changes of section of the pipe, and more or less substantially reducing the water hammer action in the upper parts of the conduit. In general the Allevard experiments have shown that the Allievi theory is correct as far as the first wave is concerned, but that the following waves, probably modified by the viscosity of the water, friction against the pipe walls, and interference on the part of other waves, direct or reflected, tend to become rapidly sinusoidal, even though the first wave following the sudden closure of the conduit outflow opening was trapezoidal.

Theory of Water Hammer (*Théorie du coup de bélier*, L. Allievi, *Bulletin technique de la Suisse Romande*, vol. 39, nos. 11 and 14, pp. 121 and 159, June 10 and July 25, 1913. Serial article, not finished. *tA*). Continuation of the abstract published in *The Journal*, August 1913, p. 1287. Let us assume that in a conduit, the permanent operation of which is characterized by y_o and v_o, a period of disturbance is created by manipulating the pipe closer, the beginning of this manipulation coinciding with the time $t=0$; by t let us denote some instant between 0 and μ, belonging, through being in that interval, to the phase of the positive blow; let us further provide with indexes:

	1	2	3	4	etc.
the values of the variable quantities corresponding to the instants					
	t_1	$t_1+\mu$	$t_1+2\mu$	$t_1+3\mu$	etc.
of the	1st	2d	3d	4th	etc. phase,

and let us denote by capitals the variations occurring at the section of the abscissa $x=0$, adjacent to the pipe closer. Equations [2] and [3] (*The Journal*, August 1913, pp. 1287–1288) can then be written as follows:

$$\left.\begin{aligned} Y_1 &= y_o + F_1 \\ Y_2 &= y_o + F_2 - F_1 \\ Y_3 &= y_o + F_3 - F_2 \\ &\ldots\ldots \end{aligned}\right\} \qquad [4]$$

$$\left.\begin{aligned} V_1 &= v_o - \frac{g}{a} F_1 \\ V_2 &= v_o - \frac{g}{a} (F_1 + F_2) \\ V_3 &= v_o - \frac{g}{a} (F_2 + F_3) \\ &\ldots\ldots \end{aligned}\right\} \qquad [5]$$

The first general consequence that can be deduced from the nature of the systems of equations [4] and [5] is that the series of load values Y_1, Y_2, Y_3, etc., as well as the corresponding series of velocity values V_1, V_2, V_3, V_4, etc., separated from one another by intervals of time equal to the duration of a phase, constitute *interconnected series*, i. e. series of values which depend only on the initial conditions and the positions of the pipe closer at the instants: t_1, $t_1+\mu$, $t_1+2\mu$, etc., but neither on the intermediary positions of the pipe closer, nor on the values of Y and V through which they may have passed in the intervals separating these instants.

The analytical expression of this series formation is obtained by eliminating F_i from the systems [4] and [5]. To do this, add each equation of [4] to the one preceding it and subtract each equation [5] from the one preceding it. The following is then obtained:

$$\left.\begin{aligned} Y_1-y_o&=\frac{a}{g}\,(v_o-V_1) \\ Y_1+Y_2-2y_o&=\frac{a}{g}\,(V_1-V_2) \\ &\ldots\ldots \end{aligned}\right\} \qquad [6]$$

If we denote by ψ the ratio between the variable area of the outflow opening, and the area of the section of the conduit, then:

$$\psi_o=\frac{v_o}{u_o}, \quad \text{and} \quad \psi_i=\frac{V_i}{u_i}$$

where u_o and u_i are the efflux velocities corresponding to loads y_o and Y_i; further, η may denote the ratio $\dfrac{\psi}{\psi_o}$, or the ratio of the actual opening of the outflow orifice, to that opening of it ψ_o which is taken as a unit.

Assume further that

$$y_o=\frac{u_o^2}{2g}; \quad Y_i=\frac{u_i^2}{2g}; \quad V_i=\eta_i u_i\frac{v_o}{u_o},$$

and introduce into the equation the characteristic ρ which may be defined by the equation:

$$\rho=\frac{av_o}{2gy_o}=\frac{av_o}{u_o^2} \qquad [7]$$

the following is then obtained:

$$\left.\begin{aligned} u_1^2-u_o^2&=2\rho u_o\,(u_o-\eta_1u_1) \\ u_1^2+u_2^2+2u_o^2&=2\rho u_o\,(\eta_1u_1-\eta_2u_2) \\ &\ldots\ldots \end{aligned}\right\} \qquad [8]$$

which is a system of quadratic equations where the only unknown magnitudes are the velocities of outflow u. The author, in accordance with the first of his principles, proceeds now to substitute for the absolute values of u relative values referred to u_o, which can be done by dividing all the equations of the system [8] by u_o^2, if $\zeta_i=\dfrac{u_i}{u_o}$, hence:

$$\left.\begin{aligned} \zeta_1^2-1&=2\rho\,(1-\eta_1\zeta_1) \\ \zeta_1^2+\zeta_2^2-2&=2\rho\,(\eta_1\zeta_1-\eta_2\zeta_2) \\ &\ldots\ldots \end{aligned}\right\} \qquad [9]$$

which connects the values of the interconnected series ζ_1, ζ_2, etc., and which the author calls *fundamental system* because, as shown later, it contains the entire

theory of the water hammer. The first of the equations of [9] can evidently be expressed in the form

$$\zeta_o^2+\zeta_1^2-2=2\rho\ (\eta_o\zeta_o-\eta_1\zeta_1)$$

since $\eta_o=1$, and $\zeta_o=1$, so that the fundamental system may be considered as resulting from the repeated application of the general equation

$$\zeta_{i-1}^2+\zeta_i^2-2=2\rho\ (\eta_{i-1}\zeta_{i-1}-\eta_i\zeta_i)$$

to a series of instants separated from one another by intervals u. *This single equation governs all the hydrodynamic phenomena which may occur in a conduit fed from a constant level reservoir and provided at the end with an outlet opening of variable cross-section.* The name of *conduit characteristic* given to ρ is justified by the fact that equation [7] expressing it, when introduced into [9], is sufficient to supply the fundamental system [9] with all the characteristic elements of the conduit, viz.: working load and speed y_o and v_o, and diameter, thickness and elasticity of the conduit, the latter through being contained in the velocity of propagation a. The only element of the conduit not contained in ρ is its length L, which however serves only to determine $\mu=\frac{2L}{a}$, the value of which determines the rhythm of the interconnected series of values of ζ_i. It is therefore shown that the relative value of the intensity of the water hammer, which is the real object of investigation, depends not only on the laws of variation of the outflow openings (contained in η), but also on the characteristic ρ, and it becomes, consequently, important both to establish the actual significance of this characteristic, and to fix the limits of numerical values which it may have within the limits of technical application. The article is to be continued.

Internal Combustion Engineering

ECONOMIC LIMITS OF GASIFICATION OF LOW GRADE FUELS (*Wirtschaftliche Grenzen der Vergasung minderwertiger Brennstoffe*, D. Bartel. *Feuerungstechnik*, vol. 1, no. 20, p. 363, July 15, 1913. 1½ pp., *g*). The editors of the publication state in an introductory notice that they do not agree with the views expressed in the article. The author does not believe in the wide application of gas producers and gas engines in large power plants, even though the gas engine appears to convert into power a larger percentage of the heating value of the fuel than the steam turbine (24 per cent as against 20 per cent). Against the success of the gas engine militates the fact that it cannot be operated efficiently in units larger than 5,000 h.p., or say 3500 kw. The speed of large gas engines is 87 to 94 r.p.m., which is ridiculously low if compared with the speed of steam turbines (3000 r.p.m. for units under 6000 kw., and at least 1000 r.p.m. for larger units). It is therefore easy to understand why gas engine plants cost so much more than steam engine plants, the first cost of the plant being often nearly treble. (Table 1.)

Altogether the cost of installing a plant of 4500 kw. driven by gas engines is about 2,000,000 marks, and only half as much with steam turbine drive, so that the cost of generation of power in the first case will be from 15 to 20 per cent higher than in the second case *even if gas be obtained free as a by-product of another manufacture.* As a matter of fact, however, the connection between by-product manufacture and power

generation is such as to prevent efficient development of the latter, owing to variations in the power load during the day, and its almost entire falling off at night: the gas has therefore at times either to be blown off into the atmosphere, or stored in gasholders, neither of which is likely to improve the economy of the plant. The author believes therefore that the use of gas from low-grade fuels may in the future develop mainly along the lines of the gas turbine, provided it can equal the steam turbine in size and cost, and excel it in efficiency; it being however also possible that the combination of the steam turbine and gas-fired boiler may present another, somewhat devious, but practical solution of the same problem. The author attaches great importance, with regard to the future field of application of gas, to the experiments of Caro and Frank on the production of ammonium sulphate, and of Asmus Jabs on by-products of tar, which, if successful, would reduce the cost of gas so materially as to open to it entirely new fields of application.

Mechanics

EXPERIMENTS ON SHOCK ABSORPTION OF POWER VEHICLES (*Versuche über die Abfederung von Kraftfahrzeugen*, E. Bobeth. *Der Motorwagen*, vol. 16,

TABLE 1 FIRST COST, IN MARKS, OF PLANT AS FUNCTION OF OUTPUT AND DRIVE

Output in Kw.	Steam Turbine Drive	Gas Engine Drive
750	82,000	190,000
1,000	93,000	220,000
2,500	150,000	420,000
3,500	180,000	550,000
6,000	250,000	....

no. 19, p. 467, July 10, 1913. 4 pp., 7 figs. to be continued *e*). The author fully describes and illustrates by figures the test installation for recording graphically the shocks, as well as for measuring the stresses produced on the various parts of the vehicle. The nature of the phenomenon of shock can be deduced from Fig. 2A. It is assumed that the wheel shown in the figure rolls along its path with a constant horizontal velocity v_h, and when at the point A meets an obstacle B which it has to jump over. At the instant of meeting the obstacle, the wheel has to perform a motion of rotation about the point C of the obstacle, and if the wheel is equipped with rigid tires, and the velocity v_h remains constant while it clears the obstacle, according to the parallelogram of velocities, at the instant of the contact with B there must appear a vertical velocity v_v of finite magnitude. Since, however, at the instant preceding that of striking the obstacle, the vertical velocity of the wheel was equal to zero, its vertical acceleration must be infinite. Actually, the acceleration is finite, because either the obstacle or the wheel undergoes changes of shape, and a good way to keep the vertical forces in finite limits is to use elastic tires, so as to permit

the obstacle partly to penetrate the tire: the vertical forces increase then from zero, and impart a gradually increasing vertical velocity to the mass of the axle, the wheel and axle executing an oscillatory motion, which depends on the shape of the obstacle, and is transmitted to the wagon body. In the above discussion it is assumed that the horizontal velocity v_h of the chassis and axle remains constant even while the wagon passes over an obstacle. This assumption is usually correct since the mass of the vehicle is so large that the retardation at the instant of the shock may be entirely neglected.

Fig. 2B shows one of the oscillation diagrams obtained, where the curve *a* represents the vibrations of the axle, and *b* those of the wagon frame, while *c* is a curve drawn to the same scale, and showing what the oscillations would have been if the wheel, equipped with a perfectly rigid tire, rolled over the obstruction without losing its contact with the ground. In the latter curve *A* indicates the instant of first contact between the wheel and obstruction, *A'* the instant when the wheel is vertically over the middle of the obstruction, and *A''* the instant when the wheel loses its contact with the obstruction. The shape of curve *c* is of course entirely

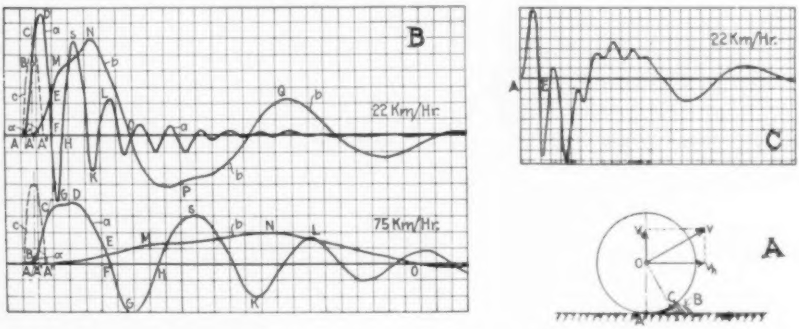


FIG. 2 SHOCK VIBRATIONS IN POWER VEHICLES

different from that of *a* showing the actual oscillations in the case of wheels equipped with elastic tires. The gradual rise of the axle is shown in the rising branch *ABC* of the curve *a*. A considerable vertical velocity is attained, which does not become zero until *D*, corresponding to the instant when the forces opposing the upward motion of the axle have entirely consumed the kinetic energy of the axle, and a downward swing of the axle begins. In this downward swing the mass of the axle acquires also a considerable amount of kinetic energy, and consequently moves past its position of equilibrium (indicated on the diagram by the horizontal line), and exerts a certain amount of pressure on the road surface *FG* until all its kinetic energy is consumed by the road resistance. The road surface pressure produces a second upward swing *GHs* followed by a series of upward and downward oscillations, gradually decreasing in amplitude; this shows that there are constantly acting forces tending to prevent the propagation of oscillatory movements. As a sequel to axle oscillations appear oscillations of the frame shown in the diagram by the curve

b; these oscillations, being of a secondary nature, lag somewhat behind the oscillations of the axle (primary), and it is of peculiar interest that not only the first, but the second upward swing of the axle as well materially affect the magnitude of the vibrations of the frame. Since the abscissae represent time, the diagram indicates directly the duration of the oscillations; while comparing the durations of oscillations of the axle and frame, it should be borne in mind that the axle is a mass oscillating between two elastic systems, the wagon springs on one hand, and the tires on the other, while the frame is supported only by one elastic system, the wagon springs; its period of oscillation is therefore usually longer than that of the axle, while with the latter the period of the first oscillation is on the average longer than that of the subsequent oscillations. The diagram affords also an easy way of determining the velocity of vibration of the axle or frame at each moment, through the fact that the velocity $v = \frac{ds}{dt}$, and is represented by the tangent to the oscillation curve.

It is further of considerable interest to determine the relative motions between the axle and frame. This may be deduced from Fig. B: the points of intersection of the curves *a* and *b* indicate moments when the axle and frame occupy mutual positions corresponding to their state of equilibrium; when the curve *a* is above the curve *b*, the axle is closer to the frame than the state of equilibrium warrants, while the location of *a* below *b* indicates their drawing wider apart. Fig. 2C represents a diagram of these relative motions, and shows that the variations of these relative positions of the axle and frame with respect to one another occurs at an extremely high speed, considerably higher than the maximum velocity of the component absolute motions. The article is to be continued.

Experimental Determination of the Coefficient of Cyclic Variation (*Die experimentelle Bestimmung des Ungleichförmigkeitsgrades*, Wilhelm Riehm, *Mitteilungen über Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, no. 137, 1913, p. 1. 32 pp., 28 figs. and *Zeits. des Vereines deutscher Ingenieure*, vol. 57, no. 28, p. 1101, July 12, 1913. 7 pp. 21 figs. *e*). The author shows that for gas engines particularly and reciprocating engines generally the theoretical velocity diagram derived from the mass-kinetic energy diagram corresponds fairly closely to the actual velocity diagram as long as there are no elastic elements in the transmissions. In practice, however, there is usually a number of influences present producing disturbances in the velocity diagram, and no calculation is possible unless the dynamic influences of all such disturbing forces are taken into account. Sometimes, as in the case of shafts subject to torsional oscillations, the production of such disturbing forces can be shown theoretically, but even there a quantitative determination is both difficult and uncertain. Generally, however, the forces producing cyclic variations cannot be determined theoretically at all, and require some instruments for their determination by actual measurement. The author describes in full an *eddy-current tachograph*, which he designed for this purpose, and which gives a diagram showing all the variations of velocity. Full data of tests for the coefficient of cyclic variations of a 10-h.p. gas engine are also given in the article.

New Method for Measuring the Velocity of Fluids (*Nouvelle méthode de mesure de la vitesse des fluides*, Z. Carrière. *Comptes rendus de l'Académie des Sciences*, vol. 156, no. 24, p. 1831, June 16, 1913, pp. *e*). Pitot tubes, anemometers and similar instruments may be made to be very sensitive, but they cannot be relied upon for measuring velocities: they have to be calibrated empirically, and this is usually done by displacing the apparatus at a given velocity in a supposedly stationary medium. The air never being perfectly calm, a correction is made for wind. This wind is supposed to be feeble, while, as has been shown (Bulletin de l'Institute Aérotechnique, fasc. II, pp. 12 and 13), the velocity in 30 seconds may vary from 1.30 to 2.90 m (4.26 to 9.51 ft.), with a change in direction up to 30 deg. Under these conditions, the correction made for the action of the wind does not correct much, and the calibration is by no means reliable. Owing to these considerations, the author worked out a *process of determining air velocities cinematically*, by incorporating into the air current, along its axis, a jet of steam under low pressure. A bundle of horizontal rays of light is then projected on the jet of steam, after being previously condensed by a crystallizer filled with water. The steam jet vibrates spontaneously, and takes the form of little clouds carried by the air. Individually these clouds are invisible, but may be conveniently observed by means of a rotating vertical mirror. The apparent trajectories of these little clouds appear then as inclined brilliant bands separated from one another by dark intervals, these bands becoming more vertical as the speed of the current of air increases, and that of rotation of the mirror diminishes. Let ϕ be the angle of the bands with the horizontal, V the velocity of the clouds, n the number of revolutions of the mirror per second, d distance from the mirror to the air current. Then

$$V = 4\pi nd \operatorname{tang}\phi$$

and since the mass of the steam is negligible when compared with the mass of the air which carries it, V soon becomes equal to the velocity of the current of air itself. The velocity of the steam jet must be neither too great nor too small, the best velocity being that of the air current itself; the steam is therefore at low pressure, and may be supplied from a glass vessel. It is as a rule not perfectly dry, but that can cause no error, since the water drops have a more horizontal trajectory, and are more brilliant than the steam. The author describes in some detail the methods of determining the angle ϕ and of producing a uniform rotation of the mirror. The principle of the process described is not new; what is new is the combination of the use of rotating vertical mirrors with the use of a material of nearly the same density as air. This process appears to be applicable only to air or gas currents having a vibration of their own, e.g., those flowing out through an orifice, in which case, as is known, there is always a feeble sound of which the height is proportional to the velocity of efflux, but independent of the size of the orifice.

Measuring Instruments

Recording Load Indicator (*Registrierender Belastungsanzeiger*, *Der praktische Maschinen-Konstrukteur*, vol. 46, no. 15, p. 67 (Section: Allgemeiner Teil), July 17, 1913. 1 p., 1 fig. *d*). It is often desirable to know

not so much the actual *load* on the engine, as its *variations during a certain period.* The apparatus here described is designed to satisfy this want. It is based on the fact that the position of the engine governor corresponds in a certain way to the load; in locomotives and marine engines the link motion does the same thing. While the principle of the apparatus is very simple, considerable constructive difficulties had to be overcome until a reliable apparatus could be produced. In Fig. 3, *c* is a heavy pendulum oscillating in a state of neutral equilibrium, and connected, by means of the springs *b*, with the driving crank *a*, in its turn connected with the governor; the springs are so selected as to take up the entire weight of the pendulum, and thus eliminate friction. To prevent the rise of resonant oscillations between the crank *a* and the pendulum, the lower end of the springs is made adjustable radially with respect to the axis of the pendulum. The apparatus shows the time of occurrence and duration of every

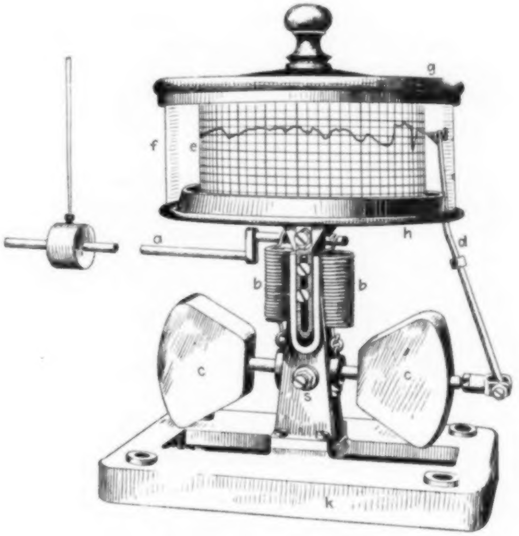


Fig. 3 Recording Load Indicator

variation in load, and may be applied to steam engines of all kinds, whether turbines or reciprocating, internal-combustion engines, hydraulic motors, etc., and is particularly recommended for tests of engine efficiencies, to show that the entire test was made at constant load. The heavy line above the recording curve indicates the curve for full load; the distance between the actual curve in each of its positions and the upper full line shows both the power reserve of the engine at each instant, and the coefficient of exploitation of the engine. The indications of the load indicator may perhaps be made still more illuminating if supplemented by a recording admission pressure gage, and a device for recording the condenser vacuum. For general purposes however the load indicator diagrams will probably be sufficient.

APPARATUS FOR MEASURING THE VIBRATIONS OF SOLID BODIES IN MOTION (*Appareil de mesure des vibrations de corps solides en mouvement*, M. Gérard. *Revue Industrielle*, vol. 44, no. 2092/27, p. 366, July 5, 1913. ½ p. *d*). Special instruments have to be used for measuring the vibrations of such elements of machinery as the chassis of an automobile or wing of an aeroplane, because there is no fixed support on which to place the registering instrument. Bourlet, working in coöperation with the Duke of Guiche and in his laboratory, has invented a suitable device for this purpose which does not require a fixed support. It consists essentially of two manometric tubes closed at one end and interconnected by a rubber tube of suitable length. To the flexible membrane of the first tube called *receiver* is attached a large and fairly heavy metal disc. This tube is attached directly to the vibrating body in such a manner that the plane of the membrane is normal to the direction of vibrations. The tube will then take part in the vibrations of the body, without disturbing them, provided its mass be small in comparison with the mass of the vibrating body. Owing to its inertia, the metal disc will develop, with respect to the tube, a relative oscillatory inverse motion, and the membrane will thus have a vibratory motion of the same periodicity as that of the main vibrating body. These vibrations are transmitted to the second tube, *recorder*, provided with a stylus registering the vibrations on a rotating cylinder. It is easy to see that the relative motion of the metal disc with respect to the manometric tube consists of: (*a*) main vibratory motion synchronous with the motion to be registered, and (*b*) secondary vibratory motion due to the elasticity of the membrane: the apparatus must of course be constructed in such a manner as to make the secondary as negligible as possible. This is attained by using a thick, well stretched membrane, and a metal disc of large diameter covering the membrane nearly completely, and leaving open only a narrow circular strip. Tests made with an apparatus constructed in accordance with these principles have shown that: (*a*) the secondary oscillatory motion is entirely negligible, in fact practically unnoticeable, so that the frequency of the vibratory motion measured and that of the stylus of the recorder are equal; (*b*) for a given frequency, the ratio $r = \frac{a}{a'}$ is constant when a is varied (a is the frequency of the oscillations to be measured, a' the frequency of the oscillation of the recording stylus); (*c*) the damping r increases when the frequency diminishes. The apparatus must therefore be calibrated so as to determine the value of r as a function of the frequency. It has been found that provided the rubber tube be not wound in too many coils, its shape does not affect the action of the apparatus, so that, within reasonable limits, it may be coiled and uncoiled.

TORSION METERS (*Über Torsionsindikatoren*, Nettmann. *Die Turbine*, vol. 9, nos. 18, 19, 20, pp. 319, 337, and 355, June 20, July 20, July 5 and 20, 1913. 12 pp., 24 figs. *d*). Brief consideration of the theory of torsion indicators, and description of their various types.

Steam Engineering

CONCERNING BLOWING-OFF OF STEAM BOILERS (*Über das Abblasen von Dampfkesseln*, W. Hopf. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 36, no. 27,

p. 327, July 4, 1913. 2 pp., 2 figs. *p*). When the boiler is fed with water which is apt to form undesirable deposits, it is an important problem to know *how to keep the contents of soluble salts in the water in the boiler so as not to let them pass a certain predetermined limit*, and so keep them from forming deposits on the boiler shell and tubes. Blowing-off a boiler is always an expensive proceeding, and must not be done oftener than is absolutely necessary. The permissible contents of salts in the boiler water is determined on the basis of various considerations, such as analysis of the water, type of boiler, demand on the boiler plant, variations of load, use of steam, and finally experience in running the particular plant. The following abstract shows how to determine when to blow-off the boiler. The notation used is: z_o contents of salts in feedwater in grams per liter (only soluble salts being here considered); z_m maximum permissible contents of salt in boiler water in grams per liter; J total boiler capacity in liters, with water up to normal level; s quantity of water evaporated in the boiler per

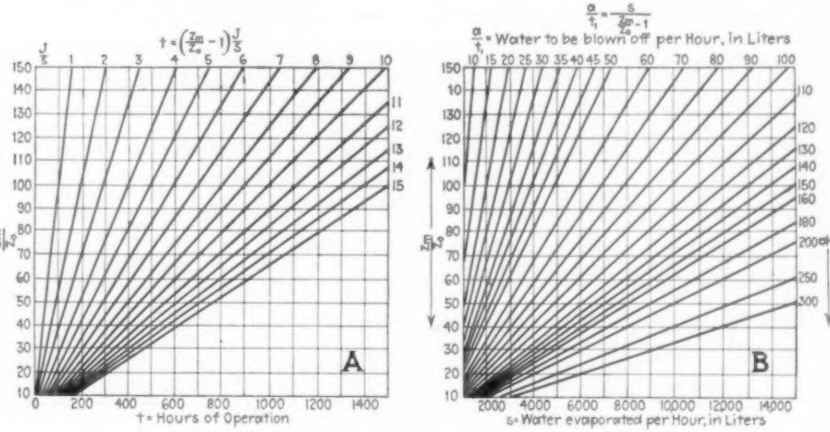


FIG. 4 BLOWING-OFF OF STEAM BOILERS

hour, in liters. The time t in hours within which the salt content of the boiler water will reach the maximum permissible amount z_m, can be determined from the following equations:

$$z_m \, . \, J = z_o \, . \, J + Z_o \, . \, s \, . \, t$$

$$t = \frac{(z_m - z_o)\, J}{z_o \, . \, s} = \left(\frac{z_m}{z_o} - 1\right)\frac{J}{s} \quad \ldots\ldots\ldots\ldots [1]$$

This shows that the time t within which the water in the boiler reaches the permissible maximum content of salts depends on the ratio $\frac{z_m}{z_o}$, as well as on the other independent variables. When the water in the boiler reaches the dangerous limit, part of it is blown-off, and the fresh water flowing in soon restores the former level in the boiler. Meanwhile the salt content in the boiler is below the permissible maximum, but rises to it gradually as the evaporation continues, and when this maximum is reached, a new blowing-off becomes necessary. If the amount a of water blown-off be expressed in liters, the periods between

blow-offs, presumably equal, may be determined from the equation (assuming, for simplicity's sake, that the blowing-off is instantaneous):

$$z_m \,.\, J - z_m \,.\, a + z_o\,(a + s \,.\, t_1) = z_m \,.\, J$$

which is true since the salt content in the boiler water at the end of the blow-off period must be equal to that at its beginning. Solving this equation for t_1 and $\frac{a}{t_1}$, the following is obtained:

$$z_m \,.\, a = z_o(a + s \,.\, t_1);\quad a\frac{z_m}{z_o} = a + s \,.\, t_1;\quad \frac{a}{t_1} = \frac{s}{\frac{z_m}{z_o} - 1} \quad \ldots\ldots\ldots\ldots [2]$$

$$t_1 = \left(\frac{z_m}{z_o} - 1\right)\frac{a}{s} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

The quantity of water to be blown off after a given time depends therefore on the amount of water evaporated in that time and on the ratio $\frac{z_m}{z_o}$, but not on the capacity of the boiler. The quantity of the water a will be established by the size and type of boiler, at the time of its construction, and equation [3] will give the best intervals for blowing off the boiler. To simplify the application of these formulae, Fig. 4A and B are given. In Fig. A $\frac{z_m}{z_o}$ indicates how many times the salt content in the boiler water exceeds that in the feedwater; $\frac{J}{s}$ shows in how many hours the entire water in the boiler will be evaporated, while t gives the number of hours in which the content of salt as present in the feedwater will be intensified $\frac{z_m}{z_o}$ times. Fig. B gives values of $\frac{a}{t_1}$ for various ratios of $\frac{z_m}{z_o}$ and quantities of steam generated per hour s, the latter from 1000 to 15,000 kg per hour. It is easy to redraw these graphs to American units, by remembering that 1 kg = 2.204 lb., and 1 liter = 0.264 gal. = 0.035 cu. ft.

Resistance to the Flow of Superheated Steam in Smooth and Corrugated Expansion Pipes (*Leitungswiderstand überhitzten Dampfes in glatten und in gewellten Ausgleichrohren*, C. Bach and R. Stücke. *Zeits. des Vereines deutscher Ingenieure*, vol. 57, no. 29, p. 1136, July 19, 1913. 8 pp., 16 figs. *e*). Data of tests made at the engineering laboratory of the Technical High School at Stuttgart, Germany. Five extension pipes were tested, two smooth and two corrugated, with and without lagging; four of these pipes had the inside diameter about 2.2 in. (55 to 56.5 mm), while one, corrugated, had an inside diameter of 4 in. (100 mm). A complete description of the method of test is given in the original. The author starts from the general formula of resistance to flow

$$H = \lambda\frac{l}{d}\,\frac{c^2}{2g}$$

where λ is an experimentally determined general coefficient of resistance to flow; l length of pipe along the axis; d inside diameter of pipe; c velocity of flow in the pipe; $g = 9.81$ (all in metric units); for the height of a liquid column H is then substituted in this formula the pressure fall $p_v = \frac{H}{v}$ 'where pressure fall p_v is re-

ferred to the sq. m. as unit, and v denotes the specific volume. The above formula, with p_v expressed in atmospheres, assumes the following form:

$$p_v = 10^{-4} \cdot \frac{\lambda}{2g} \frac{1}{v} \frac{l}{d} c^2,$$

and hence

$$10^{-4} \frac{\lambda}{2g} = p_v \, v \frac{d}{l} \frac{1}{c^2}$$

if $10^4 \frac{l}{2g}$ be denoted by β, coefficient of resistance to flow, the latter is then equal to

$$\beta = p_v \, v \frac{d}{l} \frac{1}{c^2} 10^8 \ldots\ldots\ldots\ldots [1]$$

The tests were made with superheated steam at gage pressures of 5.5 to 12.75 atmospheres at the admission opening to the expansion pipe; steam temperatures of 350 to 364 deg. cent. (662 to 687.2 deg. fahr.), and the steam velocity from approximately 51 to 121 m (167.2 to 400 ft. per sec.) (on p. 1142 the minimum steam velocity is stated to have been 25 m, but this is apparently a misprint—EDITOR). The values of the coefficient of resistance of flow have been determined and are given in Table 2.

TABLE 2 COEFFICIENTS OF RESISTANCE OF FLOW (SUPERHEATED STEAM)

Pipe No.	Diameter, Mm.	Lagging	β	Type
I	55	None	38.1	Corrugated
I	55	Some	36.2	Corrugated
II	56	None	33.6	Corrugated
II	56	Some	32.3	Corrugated
III	56.5	None	17.9	Smooth
III	56.5	Some	16.35	Smooth
IV	56.5	None	17.8	Smooth
IV	56.5	Some	16.3	Smooth
V	100	Some	49.2	Corrugated

The coefficient of resistance to flow in expansion pipe I is larger than in pipe II owing to the presence, in four places, of contractions reducing the diameter to 53 mm. The coefficient for corrugated pipe is generally about twice as large as for smooth pipe. The coefficient of resistance to flow for the large pipe (100 mm, or 4 in.) is comparatively large because the radii of curvature of the bends are small. It is worth noticing in this connection that Eberle (*Mitteilungen über Forschungsarbeiten*, no. 78, p. 65) has found that the coefficient of resistance to flow for smooth straight pipes is only 10 to 11. On the other hand the corrugated expansion pipe admits of a deflection about 5.5 times higher than a smooth one. The original article contains a complete table of data of tests.

Strength of Materials and Testing

INFLUENCE OF DRILLING OF HOLES ON THE STRENGTH OF SOFT STEELS (*Influence du perçage sur la résistance des aciers doux*, C. Birault. *Le Génie Civil*, vol. 63, no. 12, p. 230, July 19, 1913. 2 pp., 2 figs. *e*). The author, in charge of the department of testing of materials at the Ecole Centrale,

Paris, found by tests *that when holes are drilled and then reamed in soft steel bars, the metal materially increases in strength,* the average limit of elasticity improving 12.3 per cent, and the average tensile strength 9.2 per cent. The author gives the following explanation of this phenomenon. In putting together the parts of a test piece broken under tension, it is found that the two ends do not coincide; and that, while the edges make a good contact, the central parts do not, this indicating that the rupture begins at the center, and that the edges have a higher tensile resistance than there is along the axis of the bar. Therefore, if there are several holes drilled so as not to injure the material too much, as might be the case with punching, the average tensile strength of the section across the holes, per unit of metal, will be higher than before the holes were drilled, since each hole creates, so to say, additional edges. To show these differences better in the elongation of the various parts of the test bar, from axis to edge, parallel lines were drawn on a soft steel test piece (without holes), normal to its axis, and at a distance of 10 mm (say 0.2 in.) from one another. After rupture these initially parallel lines looked as shown in Fig. 5, drawn to scale. The limit of elasticity of the material was exceeded in every part of the piece at the time of rupture, and the parallel lines ceased to be parallel in every part of the bar except its ends. Near the section of rupture the lines are concave towards the rupture, elsewhere convex. In general this shows that rivet holes when properly made do not lower the tensile strength of the riveted piece in proportion to the reduction of section.

FIG. 5 DISTRIBUTION OF TENSILE STRESSES IN SOFT STEEL BARS

ELECTRICAL ATOMIZATION OF METALS FOR METALLOGRAPHIC INVESTIGATIONS (*Die elektrische Zerstäubung von Metallen zum Zweck metallographischer Untersuchungen*, G. Goldberg. *Dinglers polytechnisches Journal*, vol. 328, no. 27, p. 417, July 5, 1913. 2 pp., *dg*). Description of the Svedberg process for the atomization of metals by transferring them into a colloidal solution, for the purposes of metallographic investigations. The electrodes in the form of cylinders of about 6 mm (0.23 in.) in diameter made of the metal to be investigated are fixed in the clamps of a spark micrometer at a distance of approximately 0.25 mm (say 0.01 in.), the whole being submerged in a vessel containing some liquid of solution (ethyl ether was used by Svedberg usually, though it has been found that the nature of the liquid does not essentially affect the course of the reaction). The electrodes are connected on one hand with the secondary circuit of a Rumkorff induction coil (spark about 30 mm (1.2 in.) long, and on the other hand with a shunted-on Leyden jar of considerable capacity (say 0.0045 microfarads). Svedberg has established that for the most effective atomization the capacity must be as large, and the self-induction, ohmic resistance, and length of spark as small as possible. The colloidal solution obtained by this method may be further investigated by an ultramicroscope, while the electrode face may be submitted to the usual metallographic analysis even

though, after the atomization, it has a rather rough appearance. The action of the spark produces on the face of the electrodes a number of little craters, with generally one of larger dimensions, this being probably the starting point of the main spark discharge. These craters have different appearances for different kinds of metals, and, e.g., in the case of several test-pieces of electric steel, it has been observed that the size of the crater increases with the carbon content in the steel. The cementite contents and slag inclusions could also be easily estimated in a similar manner. The quantity of metal atomized was in all cases found to be proportional to the square of the current, and the loss in weight of the electrodes independent of the direction of the current.

Supplementary References

Breguet Ejectair (The Journal, July 1913, p 1183). Cp. Mémoires de la Société des Ingénieurs Civils de France, Ser. 7, vol. 66, no. 5, May 1913, p. 689.

Heavy Oil Kromhout Engine (The Journal, March 1913, p. 527). See also Revue industrielle, vol. 44, no. 2083/18, p. 242, May 3, 1913.

Oddie-Simplex Steam-Air Pumps (The Journal, May 1913, p. 902). Cp. Die Turbine, no. 17, p. 309.

Water Hammer (The Journal, June 1913, p. 1057). Cp. The Practical Engineer (London), vol. 47, no. 1369, pp. 402, 403, 405, and The Practical Engineer, vol. 48, no. 1379, July 31, 1913, p. 111.

NECROLOGY

WALTER S. BROWN

Walter S. Brown, who was drowned while canoeing on the Lehigh Canal, July 13, 1913, was born in St. Louis, Mo., on December 30, 1871, and received his technical education in Washington University. From 1892 to 1902 he was connected with the St. Louis waterworks, first as draftsman, and then as assistant mechanical engineer in the construction and testing department. During part of this time he served as first lieutenant of the volunteer engineering regiment in the Spanish-American War, returning from Cuba after the war in command of his company.

In 1902 he removed to Bethlehem, Pa., to become engineer in the power department of the Bethlehem Steel Works, and held this position at the time of his death. Mr. Brown was a member of several masonic and benevolent orders.

WILLIAM MASON

William Mason who until his retirement from business a few years ago was master mechanic of the Winchester Repeating Arms Company, of New Haven, died in Worcester, Mass., on July 17, 1913. He was born in Oswego, N. Y., on January 30, 1837, and early showed his natural taste for mechanics. His training for his profession was obtained through apprenticeship as a patternmaker and machinist, his first connection being with the Remington Arms Company at Ilion, N. Y. After a long association with this company he resigned to enter the Colts patent firearms works at Hartford, and later the Winchester Arms Company.

Mr. Mason was the inventor of many appliances for looms and weaving, steam pumps, bridge work, and for arms and ammunition and the machinery connected with their manufacture, and also assisted in the design and construction of the Knowles steam pump and Knowles fancy looms. He was a member of the Union League Club, New Haven, and also of a number of scientific societies.

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary, Am. Soc. M. E.

AIR BRAKE ASSOCIATION. Proc. 19th annual convention. *Boston, 1912.* Gift of the Society.

AMERICAN ASSOCIATION OF DEMURRAGE OFFICERS. Proc. 24th annual convention, 1913. Gift of association.

AMERICAN INSTITUTE OF CHEMICAL ENGINEERS. Trans. vol. 5, 1912. *New York, 1913.*

AMERICAN LABOR LEGISLATION REVIEW. Proc. 4th annual meeting. *New York, 1911.* Gift of the Society.

AMERICAN MANUFACTURERS' EXPORT ASSOCIATION. Proc. 3d annual convention, 1912. *New York, 1912.* Gift of the Society.

AMERICAN MINING CONGRESS. Proc. 15th annual session, 1912. *Denver, 1913.* Gift of congress.

AMERICAN RAILWAY BRIDGE AND BUILDING ASSOCIATION. Proc. 22d annual convention. *Chicago, 1912.* Gift of the Society.

AMERICAN ROAD BUILDERS' ASSOCIATION. Proc. 8th annual convention. *New York, 1911.* Gift of the Society.

AMERICAN SOCIETY OF INSPECTORS OF PLUMBING AND SANITARY ENGINEERS. Proc. 1912. *Pittsburgh, 1913.* Gift of the Society.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Journal, vol. 35, nos. 1-6, 1913. *New York, 1913.*

AMERICAN SOCIETY OF MUNICIPAL IMPROVEMENTS. Proc. 18th annual convention, 1911. *Milwaukee, 1911.* Gift of the Society.

ASSOCIATION FOR STANDARDIZING PAVING SPECIFICATIONS. 2d annual report, 1911. Gift of the Society.

ASSOCIATION OF CAR LIGHTING ENGINEERS. Constitution. *Chicago, 1909.* Gift of the Society.

ASSOCIATION OF ONTARIO LAND SURVEYORS. Annual report, 1912. *Toronto, 1912.* Gift of the Society.

DER BAU DER WOLKENKRATZER. Otto Rappold. *München, 1913.*

BOSTON METROPOLITAN WATER AND SEWERAGE BOARD. Annual report, 1912. *Boston, 1913.* Gift of board.

CANADIAN FORESTRY ASSOCIATION. 11th annual report, 1910. *Quebec, 1910.* Gift of the Society.

———List of Members, 1910. *Ottawa, 1910.* Gift of the Society.

CANADIAN RAILWAY CLUB. Proc. May 1912 meeting. *Montreal, 1912.* Gift of the Society.

CONGRESO CIENTIFICO (1° PAN AMERICANO). Derecho Internacional, constitucional e Historia. vol. 1, vol. 20. *Santiago de Chile*, 1912. Gift of congress.

Connecticut Society of Civil Engineers. Papers and Transactions for 1911. *New Haven, 1912.* Gift of the Society.

Eisen im Hochbau. By Stahlwerks-Verband A-G. ed. 4. *Berlin, 1913.*

Elements of Heat-Power Engineering, C. F. Hirshfeld and W. N. Barnard. *New York, 1913.*

Engineers and Architects Club of Louisville. Constitution, By-Laws and List of Members. August 1, 1911. Gift of the Society.

Engineers Club of Toronto. Proc. 1908-1909, vol. 1. *Toronto, 1909.* Gift of the Society.

Excavating Machinery. A. B. McDaniel. *New York, 1913.*

Hints on Safety Valves for Steam Boilers (Crosby). *Boston, 1913.*

History and Development of Submarine Signals, H. J. W. Fay. Gift of Submarine Signal Company.

Illinois Society of Engineers and Surveyors. Proc. 28th annual meeting, 1913. *Milwaukee, 1913.* Gift of the Society.

Iowa Engineering Society. Proc. 25th annual meeting. *Iowa City, 1913.* Gift of the Society.

Iowa Street and Interurban Railway Association. Proc. 9th annual convention, 1912. *1912.* Gift of the Society.

Kansas City Testing Laboratory. Specifications and other information concerning asphalt and other materials used in paving. bull. no. 4. *Kansas City, 1913.* Gift of laboratory.

Lakes-to-the-Gulf Deep Waterway Association. 4th annual convention. *St. Louis, 1910.* Gift of the Society.

Der Mensch und die Erde. vols. 8, 9. *Berlin.*

Modern Science Club. Year Book 1908-1909. *Brooklyn, 1909.* Gift of the Society.

National Commercial Gas Association. Proc. 8th annual convention, 1912. *New York.* Gift of association.

National Fire Protection Association. Proc. 17th annual meeting, 1913. *Boston, 1913.* Gift of association.

National Marine Engineers' Beneficial Association. Journal, January 20-24, 1913. *1913.* Gift of the Society.

National Supply and Machinery Dealers' Association. Proc. 7th annual convention. *1912.* Gift of the Society.

Natural Gas Service. S. S. Wyer. *Columbus, 1913.* Gift of author.

Necrologie. Victor Dwelshauvers-Dery. Liége. Gift of author.

New York (City) Board of Water Supply. 5th annual report. *New York, 1910.* Gift of board.

New York (City) Board of Water Supply. Report. Long Island sources. vols. 1-2. *New York, 1912.* Gift of board.

New York, New Haven and Hartford Railroad Company. General Statement. 1911, 1912. *New Haven, 1911-1912.* Gift of company.

New York Society of Architects. Year Book, 1911. *New York, 1911.* Gift of the Society.

Pocket-Book of Mechanics and Engineering. J. W. Nystrom. ed. 9. *Philadelphia, 1864.* Gift of Prof. Ira. H. Woolson.

Poor's Manual of Public Utilities, 1913. *New York, 1913.*

Poor's Manual of Railroads, 1913. *New York, 1913.*

PROBLEM OF SUBMARINE SIGNALLING. Extract from an article in National Magazine, October 1912. Gift of Submarine Signal Company.

RAILWAY STOREKEEPERS' ASSOCIATION. Report 9th annual meeting 1912. *1912.* Gift of the Society.

RAILWAY TELEGRAPH SUPERINTENDENTS. Proc. annual meeting 1912. *Milwaukee, 1912.* Gift of the Society.

SAFETY, W. H. Tolman and L. B. Kendall. *New York, 1913.*

ST. LOUIS PUBLIC LIBRARY. Annual report 1912-1913. *St. Louis, 1913.* Gift of library.

SCHIFFSTURBINEN, G. Bauer and O. Lasche. ed. 2. *München, 1913.*

SOCIETY OF CONSTRUCTORS OF FEDERAL BUILDINGS. bull., November 1912. *1912.* Gift of the Society.

SOCIETY OF ENGINEERS OF EASTERN NEW YORK. List of Members, July 15, 1911. Gift of the Society.

——Constitution and By-Laws. *Troy, 1904.* Gift of the Society.

SUBMARINE SIGNALS. *Boston, 1912.* Gift of Submarine Signal Company.

THERMODYNAMIK DER TURBOMASCHINEN, Guido Zerkowitz. *München-Berlin, 1913.*

UNIVERSITY OF NEVADA. *Register, 1912-1913. Reno, 1913.* Gift of university.

EXCHANGES

CANADIAN SOCIETY OF CIVIL ENGINEERS. Report of annual meeting, 1913, vol. 27. *1913.*

KONINK. INSTITUUT VAN INGENIEURS. Naamlijst der Leden, 1913. *1913.*

THE SHIPBUILDER. Annual international number, 1913. *Newcastle-upon-Tyne, 1913.*

UNITED ENGINEERING SOCIETY

ASSOCIATION OF IRON AND STEEL ELECTRICAL ENGINEERS CONVENTION. Structral steel poles and towers, R. Fleming. Gift of author.

CAMBRIA STEEL. Handbook of information relating to structural steel manufactured by the Cambria Steel Company. *Johnstown, 1912.* Gift of Cambria Steel Company.

CHRONOLOGY OF AVIATION, Hudson Maxim and W. J. Hammer. Reprinted from World Almanac, 1911. Gift of W. J. Hammer.

COLUMBIA UNIVERSITY. Catalogue of Officers and Graduates, 1754-1912. *New York, 1912.* Gift of university.

EFFICIENCY SOCIETY. Trans. vol. 1, 1912. *New York, 1913.* Gift of society.

INTERNATIONAL ACETYLENE ASSOCIATION. Report of 14th, 15th annual conventions. *Chicago, 1911-1912.* Gift of association.

MOODY'S MANUAL OF RAILROADS AND CORPORATION SECURITIES. vol. 2, 1913. *New York, 1913.*

QUEENS BOROUGH PUBLIC LIBRARY. Report of the chief librarian for the year ending December 31, 1912. *New York City, 1912.* Gift of library.

TRADE CATALOGUES

CHICAGO PNEUMATIC TOOL CO., *Chicago, Ill.* Cat. no. 43, Rockford railway motor cars.

CLINTON WIRE CLOTH CO., *Clinton, Mass.* Steel fabric, June 1913.

FOSTER ENGINEERING CO., *Newark, N. J.* Cat. no. 20, valve specialities of the highest order, 1913.

GOLDSCHMIDT THERMIT CO., *New York, N. Y.* Reactions, 2d quarter, 1913.

GREENE, TWEED & CO., *New York, N. Y.* Rochester automatic lubricators.

NEWALL, G. M., ENGINEERING CO., *Philadelphia, Pa.* The Ad-vance, July 1913.

NORTH WESTERN EXPANDED METAL CO., *Chicago, Ill.* Expanded metal construction, August 1913.

SPRAGUE ELECTRIC WORKS, *New York, N. Y.* Bull. 246, motor driven exhaust fan outfits.

UNDER-FEED STOKER CO. OF AMERICA, *Chicago, Ill.* Publicity magazine, July 1913.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing positions for its members. The Secretary gives this his personal attention and is pleased to receive requests both for positions and for men. Notices are not repeated except upon special request. Names and records, however, are kept on the current office list three months, and if desired must be renewed at the end of such period. Copy for the Bulletin must be in hand before the 12th of the month. The published list of "men available" is made up from members of the Society. Further information will be sent upon application.

POSITIONS AVAILABLE

714 Editor, aggressive, energetic young man of technical training, familiar with publishing business. Salary \$25-\$40 a week, depending upon experience. Location, New York.

721 Technical graduate for sales department. Prefer if possible a man who has had a few years' experience with engines or boilers. Good chance for advancement for the right man. Location, Indiana.

722 Designer and engineer having experience with four-valve engine design, construction and operation; 25-35 years of age. Salary \$1500 to \$1800. Location, New York State.

723 Partner, who can furnish necessary capital and who is successful salesman of boilers, to assist in development of a new water tube boiler of unique design, capable of good efficiency in all sizes from 100 h.p. upward; especially adapted to mill and blast furnace, central station and other operations requiring large units; some installations now operating in highly satisfactory manner. Ready for development on a large scale and now estimating on other installations.

724 Man with executive ability and good knowledge of building work of all kinds, both in steel, brick and reinforced concrete, versed in mechanical, electrical engineering and hydraulics.

725 Engineer to operate steam railroad, electric power plant, saw mill, logging machinery, etc. Location, North Carolina. Salary \$250-\$300.

726 New York concern desires to establish additional agencies in the following cities: Concord, N. H.; Burlington or Rutland, Vt.; Boston, Mass.; Springfield or Worcester, Mass.; Albany, N. Y.; Utica, N. Y.; Syracuse, N. Y.; Rochester, N. Y.; Trenton, N. J.; New Haven, Conn.; Hartford, Conn.; Harrisburg, Pa.; Scranton, Pa.; Providence, R. I.; Washington, D. C. These agencies could be handled by men representing other concerns.

801 Member, or member's son, with business ability and \$10,000 can secure equal partnership with full financial control and management in new concern, to engage in the manufacture of a complete line of machinery fully covered by patents. Sales in the last three years, \$100,000.

802 Large eastern university desires five instructors in mechanical engineering and mechanical engineering laboratory; three instructors in elec-

trical and electrical engineering laboratory and one instructor in drawing and design.

803 A man of experience and ability to organize a repair department of about 150 men; in charge of machinery in a plant employing about 800 men and representing an investment of between three and four million dollars. Salary $4000. Location Middle West. Apply through the Society.

804 Engineer with experience in heat, light and power work; 35 to 40 years of age. Salary $2500-$3000. Location, Philadelphia. Apply through the Society.

805 Instructor in mechanical engineering for one of the larger engineering institutions. Practical experience in the field. Salary $1300 for year of 12 months.

806 Instructor in machine shop practice wanted in western university school of engineering. Technical graduate preferred, but not required. Shop experience necessary. Will be required to instruct in modern shop methods, cost systems, standardized production, etc. Some time to be devoted to instrument making. Salary $1200 or more to man of wide experience.

MEN AVAILABLE

193 Sales engineer desires position where a knowledge of machinery and mill supply trade in United States and Canada is essential; seven years varied experience, nine years in selling end. Experience in correspondence and design of selling contracts.

194 Junior member, age 25, technical graduate, Columbia University, 1911, desires position either as assistant superintendent in factory, construction work, or in the heating and ventilating line. Broad experience as factory inspector.

195 Member, technical graduate, desires to secure position as assistant to consulting engineer or with manufacturing concern as mechanical engineer and efficiency expert. At present employed. Apply through the Society.

196 Young engineer, graduate Stevens Institute of Technology, experienced in electrical engineering in consulting engineer's office. Desires to change for prospect of advanced work in mechanical lines.

197 Member, age 31, married, well informed and of good address, desires change. Thorough operating and commercial experience. College trained. Assistant to executive, administrative or general sales officer preferred. Salary $3600.

198 Junior member, age 32, technical graduate in mechanical engineering, experienced in detailing, designing and estimating on steel plate work, familiar with boiler shop work, for last six years assistant manager, wishes a position with a company where such experience could be used.

199 Junior member, age 32, graduate in mechanical engineering, experienced in estimating and designing springs for heavy work and railroads, wishes position with a spring manufacturing company. Has also had extensive office experience as assistant manager in other lines.

200 Member, long experience on light and medium light weight manufacturing, desires position as works manager or general superintendent.

201 Member, 20 years varied experience in designing, building and

operating. At present in charge of large construction work nearing completion, also surface equipment of two mines and gravel plant on Pacific Coast. Has made special study of economical methods of handling materials. Accustomed to handling men, organizing crews, drawing up contracts, designing and purchasing; desires connection with some firm handling large work.

202 Junior member, age 29, extensive experience here and abroad in designing and operating by-product coke ovens, first-class organizer and expert furnaceman (Bunteschueler). Desires position as operating engineer or superintendent.

203 Member, master's degree from Cornell, desires position as electrical, mechanical or efficiency engineer, purchasing agent or manager of an industrial plant. Has had 20 years experience designing, constructing, operating and managing. Can give the best of references.

204 Junior member, age 26, technical graduate, at present connected with large hydroelectric project nearing completion, desires position in power or industrial plant design. Several years experience. References.

205 Student member, age 23, technical graduate with shop experience of a varied nature, has worked as draftsman and as designer; now assistant chief engineer in a boiler and engine concern of wide reputation. Salary depends upon opportunity. Reference can be given.

206 Mechanical engineer, technical graduate, age 35, with broad experience in central station work, designing, buying equipment, selling old material, supervision of installation, management of operation, in the construction and operating department; investigating requirements of large power consumers, estimating on cost to deliver service, management of isolated plants and the heating of buildings in the commercial department work. Desires position as manager or chief engineer. At present employed.

207 Member, 20 years experience in shop and office, last 5 years confidential aid to consulting engineer, having general charge of drafting room and design of railway and lighting power plants, special apparatus, etc., handling reports, specifications and correspondence, wants responsible position as engineer with operating company, manufacturing concern or consulting engineer.

208 Mechanical engineer, age 29, graduate Mass. Inst. of Tech., excellent experience in design of industrial plants and mechanical equipment of buildings, desires permanent connection with firm of consulting or mill engineers or position as plant engineer.

209 Member, age 38, technical graduate, desires position as assistant professor in an engineering school, preferably in the East. Eleven years practical experience, three years teaching and two years recent foreign study.

210 Mechanical engineer, technical graduate, German, age 33, at present shop superintendent for Diesel motor construction, would like similar position in U. S. or Canada. Has 13 years experience.

211 Member, technical graduate, age 34, American, capable engineer, practical foundry and shop man, desires position as principal or assistant engineer, superintendent or manager. Broad engineering experience in

Europe and America as mechanic and executive in shop and plant construction, operation and maintenance, heavy machine building and light high grade manufacture. Salary commensurate with position.

212 Member, 31 years of age, at present employed, technical education, 10 years experience in design, construction, operation, maintenance and reorganization of mill, factory and other manufacturing properties. Wide experience in the superintendence of central power stations, factory extension, mill and reinforced concrete construction work.

213 Engineer, eight years experience in mechanical, electrical and construction work, desires a position leading to one of responsibility.

214 Member, age 54, has had charge of the design and equipment of the locomotive and car repair shops of three large railroads; also of power house work for trolley lines; for five years superintendent of bridges and buildings for a large eastern railroad.

215 Junior member; married, desires position as superintendent or business manager, preferably with an educational or medical institution. Seven years in present position as assistant manager of an important New York institution. Experienced in employing, organizing, purchasing, planning and supervision of building construction. Also five years of practical engineering experience. Salary dependent upon opportunity. Very best of references.

216 Graduate mechanical engineer, age 36, desires permanent position in or near New York. Excellent experience in engineering, purchasing and sales work with consulting, manufacturing and selling concerns. At present employed in executive sales position, but desires to change for better and more permanent work.

217 Factory manager or superintendent thoroughly familiar with modern machine shop practice, and production, capable of taking full charge of manufacturing plant, open for engagement after September 15.

218 Junior member, age 24, desires position as engineer, consulting or designing preferred. Experience mostly practical.

219 Member, graduate mechanical engineer, now employed, desires change; 15 years experience in design, construction and maintenance of power and heating plants, industrial plant equipment and cost estimating; also familiar with design and manufacture of automobiles, transmission and hoisting machinery; several years shop training. Chief designer factory engineer or superintendent; would also consider teaching position with large engineering school.

OFFICERS AND COUNCIL

President

W. F. M. Goss

Vice-Presidents

Terms expire 1913	Terms expire 1914
Wm. F. Durand	James Hartness
Ira N. Hollis	I. E. Moultrop
Thos. B. Stearns	H. G. Stott

Managers

Terms expire 1913	Terms expire 1914	Terms expire 1915
D. F. Crawford	Chas. J. Davidson	W. B. Jackson
Stanley G. Flagg, Jr.	Henry Hess	H. M. Leland
E. B. Katte	Geo. A. Orrok	Alfred Noble

Past-Presidents

Members of the Council for 1913

M. L. Holman
Jesse M. Smith
Alex. C. Humphreys
George Westinghouse
E. D. Meier

Chairman of Finance Committee

Robert M. Dixon

Treasurer

William H. Wiley

Honorary Secretary

F. R. Hutton

Secretary

Calvin W. Rice

Executive Committee of the Council

W. F. M. Goss, *Chmn.*
Alex. C. Humphreys, *V-Chmn.*
E. B. Katte
E. D. Meier
Geo. A. Orrok
H. G. Stott

STANDING COMMITTEES

Finance

R. M. Dixon (1), *Chmn.*
W. H. Marshall (2)
H. L. Doherty (3)
W. L. Saunders (4)
W. D. Sargent (5)

Meetings

H. de B. Parsons (1), *Chmn.*
L. P. Alford (2)
H. E. Longwell (3)
H. L. Gantt (4)
R. H. Fernald (5)

Publication

G. I. Rockwood (1) *Chmn.*
G. M. Basford (2)
C. I. Earll (3)
I. E. Moultrop (4)
F. R. Low (5)

Membership

Hosea Webster (1), *Chmn.*
Theodore Stebbins (2)
W. H. Boehm (3)
H. C. Meyer, Jr. (4)
L. R. Pomeroy (5)

Library

L. Waldo (4), *Chmn.*
C. L. Clarke (1)
Alfred Noble (2)
W. M. McFarland (3)
The Secretary

Research

R. H. Rice (4), *Chmn.*
L. S. Marks (1)
A. L. De Leeuw (2)
R. C. Carpenter (3)
R. D. Mershon (5)

House

E. Van Winkle (1), *Chmn.*
H. R. Cobleigh (2)
S. D. Collett (3)
W. N. Dickinson (4)
F. A. Scheffler (5)

Public Relations

J. M. Dodge (3), *Chmn.*
J. W. Lieb, Jr. (1)
F. J. Miller (2)
W. R. Warner (4)
G. M. Brill (5)

SOCIETY REPRESENTATIVES

Engineering Education

A. C. Humphreys
F. W. Taylor

General Conference Committee of National Engineering Societies

C. W. Baker
E. D. Meier

Council of A. A. A. S

A. C. Humphreys
W. B. Jackson

John Fritz Medal

H. R. Towne (1)
J. A. Brashear (2)
F. R. Hutton (3)
J. R. Freeman (4)

Trustees U. E. S.

A. C. Humphreys (1)
F. J. Miller (2)
Jesse M. Smith (3)

SPECIAL COMMITTEES

Research Committee, Sub-Committee on Steam

R. H. Rice, *Chmn.*
C. J. Bacon
E. J. Berg
W. D. Ennis
L. S. Marks
J. F. M. Patitz

Power Tests

D. S. Jacobus, *Chmn.*
G. H. Barrus, *V-Chmn.*
E. T. Adams
L. P. Breckenridge
William Kent
E. F. Miller
Arthur West
A. C. Wood

Student Branches

F. R. Hutton, *Chmn.*
Wm. Kent
Geo. A. Orrok

Research Committee. Sub-Committee on Materials of Electrical Engineering

R. D. Mershon

Note—Numbers in parentheses indicate number of years the member has yet to serve.

SPECIAL COMMITTEES—*Continued*

Committee to Formulate Standard Specifications for the Construction of Steam Boilers and other Pressure Vessels and for Care of Same in Service

J. A. Stevens, *Chmn.*
W. H. Boehm
R. C. Carpenter
Richard Hammond
C. L. Huston
H. C. Meinholtz
E. F. Miller

Tolerances in Screw Thread Fits

L. D. Burlingame, *Chmn.*
Elwood Burdsall
F. G. Coburn
F. H. Colvin
A. A. Fuller
James Hartness
H. M. Leland
W. R. Porter
F. O. Wells

Conference Committee on Engineering Standards

H. G. Stott, *Chmn.*
A. F. Ganz
Carl Schwartz

Society History

J. E. Sweet, *Chmn.*
F. R. Hutton, *Secy*
H. H. Suplee

Committee on International Engineering Congress 1915

The President
The Secretary
W. F. Durand
R. S. Moore
T. W. Ransom
C. R. Weymouth

Pipe Threads

E. M. Herr, *Chmn.*
W. J. Baldwin
G. M. Bond
S. G. Flagg, Jr.

L. V. Benet, *Paris Rep.*

Flanges

H. G. Stott, *Chmn.*
A. C. Ashton
A. R. Baylis
W. M. McFarland
Wm. Schwanhausser
J. P. Sparrow

Committee on Bureau of Information Respecting Engineering Standards in all Countries

Henry Hess, *Chmn.*
J. H. Barr
Charles Day
C. J. Davidson
Carl Schwartz

Research Committee. Sub-Committee on Safety Valves

P. G. Darling, *Chmn.*
H. D. Gordon
E. F. Miller
F. L. Pryor
F. M. Whyte

Committee on National Museum

E. D. Meier, *Chmn.*
G. F. Kunz
George Mesta
H. G. Reist
Ambrose Swasey

Standard Cross-Section Symbols

H. deB. Parsons, *Chmn.*
F. deR. Furman
A. E. Norton
Bradley Stoughton
John W. Upp

Changes in the Patent Laws of U. S.

W. H. Blauvelt
B. F. Wood
Carl Thomas
Edward Weston
W. E. Winship

Code of Ethics

C. W. Baker, *Chmn.*
C. T. Main
E. D. Meier
Spencer Miller
C. R. Richards

Refrigeration

D. S. Jacobus, *Chmn.*
P. De C. Ball
E. F. Miller
A. P. Trautwein
G. T. Voorhees

Involute Gears

Wilfred Lewis, *Chmn.*
Hugo Bilgram
E. R. Fellows
C. R. Gabriel
C. G. Lanza

Nominating Committee

J. R. Freeman, *Chmn.*
Chas. T. Main
Thos. Morrin
Fred. Sargent
C. C. Thomas

Conservation

G. F. Swain, *Chmn.*
C. W. Baker
L. D. Burlingame
M. L. Holman
Calvin W. Rice

Standardization of Filters

G. W. Fuller, *Chmn.*
Jas. C. Boyd
P. N. Engel
J. C. W. Greth
Wm. Schwanhausser

Standardization of Catalogues

Wm. Kent, *Chmn.*
J. R. Bibbins
M. L. Cooke
W. B. Snow

SPECIAL COMMITTEES—*Continued*

Chairmen of Sub-Committees of Committee on Increase of Membership

Boston, A. L. Williston
Buffalo, W. H. Carrier
Chicago, Fay Woodmansee
Cincinnati, J. T. Faig
Michigan, H. W. Alden
New York, J. A. Kinkead
Philadelphia, T. C. McBride
St. Louis, John Hunter
St. Paul, Max Toltz
San Francisco, Thomas Morrin
Seattle, R. M. Dyer
Troy, A. E. Cluett

Increase of Membership

I. E. Moultrop, *Chmn.*
F. H. Colvin
H. V. O. Coes
J. V. V. Colwell
R. M. Dixon
W. R. Dunn
J. P. Ilsley
E. B. Katte
R. B. Sheridan
H. Struckmann

SUB-COMMITTEES OF THE COMMITTEE ON MEETINGS

Cement Manufacture

J. G. Bergquist, *Chmn.*
H. J. Seaman, *V-Chmn.*
G. S. Brown
W. R. Dunn
F. W. Kelley
Morris Kind
F. H. Lewis
W. H. Mason
R. K. Meade
Ejnar Posselt
H. Struckmann
A. C. Tagge
P. H. Wilson

Machine Shop Practice

F. H. Colvin, *Chmn.*
E. P. Bullard
L. D. Burlingame
W. L. Clark
A. L. DeLeeuw
W. H. Diefendorf
F. L. Eberhardt
F. A. Errington
A. J. Fuller
H. D. Gordon
H. K. Hathaway
Alex. Kearney
Wm. Lodge
F. E. Rogers
N. E. Zusi

Textiles

C. T. Plunkett, *Chmn.*
E. W. Thomas, *Secy.*
D. M. Bates
John Eccles
E. D. France
E. F. Greene
F. W. Hobbs
C. R. Makepeace
C. H. Manning
H. F. Mansfield

Iron and Steel

Jos. Morgan, *Chmn.*
Thos. Towne, *Secy.*
W. P. Barba
F. F. Beall
Rogers Birnie
A. L. Colby
Julian Kennedy
M. T. Lothrop
W. E. Snyder
J. T. Wallis
R. M. Watt

Depreciation and Obsolescence

Alex. C. Humphreys, *Chmn.*
J. G. Bergquist
C. J. Davidson
A. E. Forstall
F. W. Kelley
H. Struckmann

Air Machinery

F. W. O'Neil, *Chmn.*
H. V. Conrad
William Prellwitz
R. H. Rice

Administration

J. M. Dodge, *Chmn.*
L. P. Alford, *Secy.*
D. M. Bates
John Calder
H. A. Evans
James Hartness
W. B. Tardy
Alexander Taylor
H. H. Vaughan

Industrial Building

F. A. Waldron, *Chmn.*
H. A. Burnham
Charles Day
William Dalton
J. O. DeWolf
C. T. Main

Railroads

E. B. Katte, *Chmn.*
G. M. Basford
W. G. Besler
A. H. Ehle
T. N. Ely
W. F. M. Goss
A. L. Humphreys
W. F. Kiesel
W. B. Potter
N. W. Storer
H. H. Vaughan
R. V. Wright

Fire Protection

J. R. Freeman, *Chmn.*
E. V. French, *V-Chmn.*
Albert Blauvelt
F. M. Griswold
H. F. J. Porter
T. W. Ransom
I. H. Woolson

Hoisting and Conveying

R. B. Sheridan, *Chmn.*
C. K. Baldwin
Alex. C. Brown
O. G. Dale
P. J. Fickinger
F. E. Hulett
Spencer Miller
A. L. Roberts
Harry Sawyer

GEOGRAPHICAL SECTIONS OF THE SOCIETY

Milwaukee

Fred H. Dorner, *Chmn*
M. A. Beck
Arthur Simon
Henry Weickel
E. P. Worden

Cincinnati

A. L. DeLeeuw, *Chmn.*
J. T. Faig, *Secy.*
W. G. Franz
G. W. Galbraith
L. H. Thullen

St. Louis

E. L. Ohle, *Chmn.*
F. E. Bausch, *Secy.*
F. N. Jewett
John Hunter
L. C. Nordmeyer

San Francisco

A. M. Hunt, *Chmn.*
T. W. Ransom, *Secy.*
W. F. Durand
E. C. Jones
Thos. Morrin

LOCAL MEETINGS OF THE SOCIETY

Boston

Henry Bartlett, *Chmn.*
R. E. Curtis, *Secy.*
H. N. Dawes
W. G. Snow
A. L. Williston

New York

F. A. Waldron, *Chmn.*
Edward Van Winkle, *Secy.*
R. V. Wright, *Treas.*
H. R. Cobleigh
J. J. Swan

Chicago

Paul P. Bird, *Chmn*
C. W. Naylor, *Secy.*
C. R. Birdsey
W. B. Jackson
A. W. Moseley

Philadelphia

A. C. Jackson, *Chmn.*
D. R. Yarnall, *Secy.*
J. E. Gibson
W. C. Kerr
T. C. McBride

New Haven

E. S. Cooley, *Chmn.*
E. H. Lockwood, *Secy.*
F. L. Bigelow
L. P. Breckenridge
H. B. Sargent

OFFICERS OF THE GAS POWER SECTION

Chairman

F. R. Hutton

Secretary

Geo. A. Orrok

Gas Power Executive Committee

C. H. Benjamin (1)
W. H. Blauvelt (3)
W. D. Ennis (5)
H. J. K. Freyn (1)
F. R. Low (2)
I. E. Moultrop (4)
H. H. Suplee (1)

Gas Power Literature Committee

R. B. Bloemeke, *Chmn.*
A. W. H. Griepe
W. F. Monaghan
W. S. Morrison
S. I. Oesterreicher
H. G. Wolfe

Gas Power Membership Committee

A. F. Stillman, *Chmn.*
H. V. O. Coes
J. H. Lawrence
F. S. King
J. H. Norris
G. M. S. Tait
J. D. Shaw
H. W. Anderson
C. D. Smith

Gas Power Committee on Meetings

Wm. T. Magruder, *Chmn.*
W. H. Blauvelt
E. D. Dreyfus
A. H. Goldingham
Nisbet Latta
H. B. MacFarland

OFFICERS OF AFFILIATED SOCIETY

Providence Association of Mechanical Engineers

T. M. Phetteplace, *Pres.*
J. A. Brooks, *Secy.*
W. H. Paine, *Vice-Pres.*
A. H. Whatley, *Treas.*

Note—Numbers in parentheses indicate number of years the member has yet to serve

OFFICERS OF THE STUDENT BRANCHES

INSTITUTION	DATE AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	CHAIRMAN	CORRESPONDING SECRETARY
Armour Inst. of Tech.	Mar. 9, 1909	G. F. Gebhardt	H. E. Erickson	A. N. Koch
Case School of Applied Science	Feb 14, 1913	F. H. Vose	H. C. Mummert	C. Stemm
Columbia University	Nov. 9, 1909	Chas. E. Lucke	F. B. Schmidt	H. F. Allen
Cornell University	Dec. 4, 1908	R. C. Carpenter	J. G. Miller	Edw. Mendenhall
Lehigh University	June 2, 1911	P. B. de Schweinitz	W. C. Owen	T. G. Shaffer
Leland Stanford Jr. Univ.	Mar. 9, 1909	W. F. Durand	C. T. Keefer	K. J. Marshall
Mass. Inst. of Tech.	Nov. 9, 1909	E. F. Miller	W. H. Treat	L. L. Downing
New York University	Nov. 9, 1909	C. E. Houghton		
Ohio State University	Jan.10, 1911	Wm. T. Magruder	R. H. Neilan	R. M. Powell
Penna. State College	Nov. 9, 1909	J. P. Jackson	H. L. Swift	H. L. Hughes
Poly. Inst. of Brooklyn	Mar. 9, 1909	W. D. Ennis	B. L. Huestis	A. Bielek
Purdue University	Mar. 9, 1909	G. A. Young	A. D. Meals	G. F. Lynde
Rensselaer Poly. Inst.	Dec. 9, 1910	A. M. Greene, Jr.	E. Kneass	R. F. Fox
State Univ. of Iowa	Apr.11, 1913	R. S. Wilbur	F. H. Guldner	C. S. Thompson
State Univ. of Kentucky	Jan.10, 1911	F. P. Anderson	R. R. Taliaferro	F. J. Forsyth
Stevens Inst. of Tech.	Dec. 4, 1908	Alex.C.Humphreys	L. F. Bayer	C. H. Colvin
Syracuse University	Dec. 3, 1911	W. E. Ninde	O. W. Sanderson	R. A. Sherwood
Univ. of Arkansas	Apr.12, 1910	B. N. Wilson	M. McGill	C. Bethel
Univ. of California	Feb.13, 1912	Joseph N. LeConte	J. F. Ball	G. H. Hagar
Univ. of Cincinnati	Nov. 9, 1909	J. T. Faig	A. O. Hurxthal	E. A. Oster
Univ. of Illinois	Nov. 9, 1909	W. F. M. Goss	A. H. Aagaard	H. E. Austin
Univ. of Kansas	Mar. 9, 1909	F. W. Sibley	E. A. Van Houten	L. E. Knerr
Univ. of Maine	Feb. 8, 1910	Arthur C. Jewett	E. H. Bigelow	O. H. Davis
Univ. of Missouri	Dec. 7, 1909	H. Wade Hibbard	W. P. Jesse	R. Runge
Univ. of Minnesota	May 12, 1913			
Univ. of Nebraska	Dec. 7, 1909	J. D. Hoffman	A. A. Luebs	G. W. Nigh
Univ. of Wisconsin	Nov. 9, 1909	A. G. Christie	W. K. Fitch	J. W. Griswold
Washington University	Mar.10,1911	E. L. Ohle	D. Southerland	A. Schleiffarth
Yale University	Oct.11, 1910	L. P. Breckenridge	C. E. Booth	O. D. Covell

APPLICATIONS MUST BE FILED BY SEPTEMBER 25

The season of the Society's greatest activity is about to begin, and this year a program has been planned that will be of great value and interest.

Those who are desirous of participating in the privileges which the Society offers should file their applications not later than September 25 for the following reasons:

a To participate as a member at the Annual Meeting in December.

b To be included in the Annual Year Book of the Society.

c To secure the published proceedings of the meetings during 1913.

d To participate in the frequent meetings held in the principal cities of the country, thereby increasing one's acquaintanceship, professional knowledge, etc.

Applications received after September 25 cannot be acted upon until after the Annual Meeting and will not be included in the 1914 issue of the Year Book.

Applications and information regarding membership may be obtained from the Secretary or from any of the undersigned.

COMMITTEE ON INCREASE OF MEMBERSHIP

I. E. Moultrop, *Chairman*

H. V. O. Coes	R. M. Dixon	E. B. Katte
F. H. Colvin	W. R. Dunn	R. B. Sheridan
J. V. V. Colwell	J. P. Ilsley	H. Struckmann

Chairmen of Sub-Committees

Boston, A. L. Williston	*New York*, J. A. Kinkead
Buffalo, W. H. Carrier	*Philadelphia*, T. C. McBride
Chicago, Fay Woodmansee	*St. Louis*, John Hunter
Cincinnati, J. T. Faig	*St. Paul*, Max Toltz
Cleveland, R. B. Sheridan	*San Francisco*, Thos. Morrin
Michigan, H. W. Alden	*Seattle*, R. M. Dyer

Troy, A. E. Cluett